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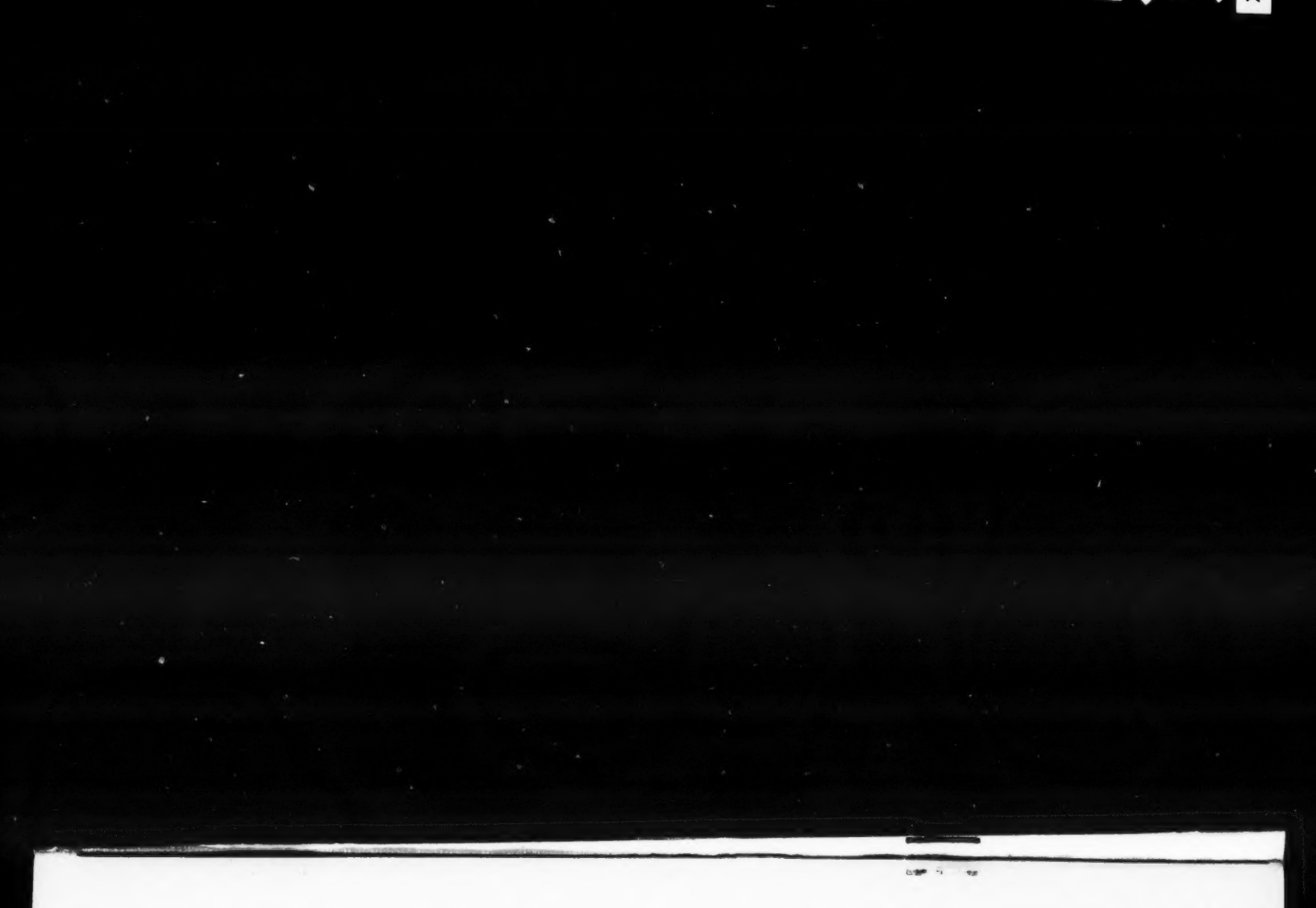

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SIXTY-SEVENTH YEAR

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I

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TABLE OF CONTENTS

PAGE

Secretary's Notes 3

Joint Council Meeting 9

Half A Century of Progress In Naval Boiler Water Treatment
Frank E. Clarke 11

Present Status of Cavitation Research
"Mechanical Engineering"—September, 1954 44

Fundamentals of Hydrodynamic Lubrication
Mayo D. Hersey 51

Engineering Communications—
What's Wrong?—Too Many Change Orders Skyrocket Costs
Captain R. E. W. Harrison, USNR 69

"Military Necessity Versus Economy"
Captain R. E. W. Harrison, USNR 75

Battle of the Boilers
"The Engineer"—July 16, 23, 1954 80

Non-Ferrous Metals and Marine Engineering
"The Journal of the Institute of Metals"
July, 1954 89

Will Nuclear Fuels Run Merchant Ships?
Holmes F. Crouch 113

An Adventure In Cooperation
R. V. Vittucci 121

Developments In Aircraft Electrical Power
"Aero Digest"—October, 1954 129

Recent Developments In Boiling Research
W. H. Jens and G. Leppert 137

Training of Engineer Officers for the Royal Navy
"The Engineer"—13 August, 1954 156

Laminar Heat Transfer In Rectangular Tubes with Combined Free-and Forced-Convection
L. S. Han 163

Shipboard Measurement of Insulation Resistance of Energized D. C. Circuits
Lt. Carvel Hall Blair, USN 169

Some Electronic Instruments for Marine Turbine Research
"Transactions of the Institution of Engineers and Shipbuilders In Scotland"—Volume 97, Part 6 175

A Practical Method for Cleaning and Descaling Cargo Tanks Contaminated With Black Oil Residues
J. F. Mills, F. E. Cook and H. S. Preiser 199

Driving Point Admittance of A Symmetrically Folded Antenna
Commander Charles W. Harrison, Jr., USN 213

The Economic Speed of Tow On Inland Waterways
M. Dimitryevic 239

Discussion 251

Book Review 253

Obituary 255

DEPARTMENTS

Deaths 256

Changes In Membership 257

Association Notes 260

Editor: Captain J. E. Hamilton, U.S.N., Ret.

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INDEX TO ADVERTISERS

Name	Manufacturers of or dealers in	Post office address	Page
A			
Allis-Chalmers Mfg. Co.	Ship Propulsion and other Equipment for Marine Use	Milwaukee, Wis.	xxix
American Brass Co.	Engineering Service on Anaconda Metals	Waterbury, Conn.	xvi
B			
Babcock & Wilcox Co.	Marine Water-Tube Boilers, Superheaters, Airheaters, Economizers, Oil Burners, Refractories, Seamless and Welded Tubing	Room 665, 161 East 42nd St., New York 17, N.Y.	xxii
Bath Iron Works	Shipbuilders and Engineers	Bath, Me.	xxi
Bethlehem Steel Co.: Shipbuilding Division	Steam Turbines, Marine Boilers, Oil Burning Equipment, Shipbuilders and Repairers	25 Broadway, New York, N. Y.	i
C			
Cleveland Diesel Engine Division, General Motors Corp.	Diesel Engines	Cleveland, Ohio	xix
Collins Radio Company	Designers and manufacturers of radio communication and navigation equipment for the Armed Services	Cedar Rapids, Iowa	xiii
Combustion Engineering Co.	Marine Steam Generators, Bent Tube and Straight Tube Boilers, Forced-Circulation Boilers, Superheaters, Economizers and Air Heaters	200 Madison Ave., New York City	iv
Cooper-Bessemer Corp. The	Marine Diesel Engines	25 West 43rd St., New York City	xxx
Crane Co.	Crane Valves	836 S. Michigan Ave., Chicago, Ill.	v
Cutler-Hammer, Inc.	Electric Control Apparatus	1354 St. Paul Ave., Milwaukee, Wis.	xiii
D			
De Laval Steam Turbine Co.	Steam Turbines	485 Nottingham Way, Trenton, N. J.	xv
E			
Electric Boat Co.	Better Submarines for Greater Security	445 Park Avenue, New York 22, N. Y.	ii
Elliott Co.	Elliott Motors, Generators, Steam and Gas Turbines, Condensers, Deaerating Feedwater Heaters, Turbo Chargers, Strainers, Tube Cleaners, etc.	Jeanette, Pa.	xiv

Name	Manufacturers of or dealers in	Post office address	Page
F			
Farrel-Birmingham Co., Inc.	Precision Gears	Ansonia, Connecticut	xvii
G			
General Electric	Electric Machinery of All Classes	Schenectady, N. Y.	iii
Gibbs & Cox, Inc.	Naval Architects and Engineers.	New York, N. Y.	xviii
Griscom-Russell Co.	Distilling Plants, Feed Water Heaters	285 Madison Ave., New York	xi
I			
International Nickel Co.	Producers of Nickel	67 Wall St., New York, N. Y.	x
J			
Johns-Manville	Insulation, Refractory Cements, Packings, Brake Linings, Asbestos Ebony for Switch and Panel Boards, Waterproofing Transite Asbestos Sheets, Transite Pipe, Roofing and Shingles, Flooring, Insulating Board Products	22 East 40th St., New York City	xiii
K			
Kingsbury Machine Works, Inc.	Thrust Bearings, Journal Bearings, Thrust Meters	Frankford, Philadelphia, Pa.	xviii
L			
Lanman Engraving Co.	Photo Engravers	612 L St., N. W., Washington, D. C.	vii
Lidgerwood Industries, Inc.	Steering and Deck Machinery	7 Dey Street, N. Y., 7, N. Y.	xxiv
M			
Moffitt, Lucian Q., Inc.	Cutless Bearings for Stern Tubes and Struts	Akron, Ohio	vi
N			
Newport News Shipbuilding & Drydock Co.	Naval and Merchant Vessels	Newport News, Va.	vii

Name	Manufacturers of or dealers in	Post office address	Page
R			
Radio Corporation of America: Govt. Equip. Section	Radio Equipment	Camden, N. J.	xxv
Rosenblatt, M. & Son	Naval Architects, Marine Engineers	111 Broadway, New York, N.Y.	xx
S			
Sperry Gyroscope Co., Inc.	Radar, Loran, Gyro-Pilots, Gyro-Compasses, Electric and Hydraulic Steering Equipments, Magnetic Compass Pilot, Rudder Angle Indicator, Naval Fire Control Instruments, Aircraft Gyropilots, Flight Instruments	Great Neck, N. Y.	xii
T			
Terry Steam Turbine Co.	Steam Turbines	Hartford, Conn.	xxviii
U			
U. S. Naval Institute	U. S. Naval Institute Proceedings	Annapolis, Md.	xxvii
United States Steel Corp.	Steel	Pittsburgh, Pa.	xxvi
W			
Westinghouse Electric & Mfg. Co.	Marine Steam and Electrical Equipment	East Pittsburgh, Pa.	viii and ix
Worthington Pump & Machinery Corp.	Pumps, Condensing Apparatus, Diesel Oil Engines, Gas Engines, Fluid Meters, Compressors, Feed Water Heaters	Harrison, N. J.	xxiii

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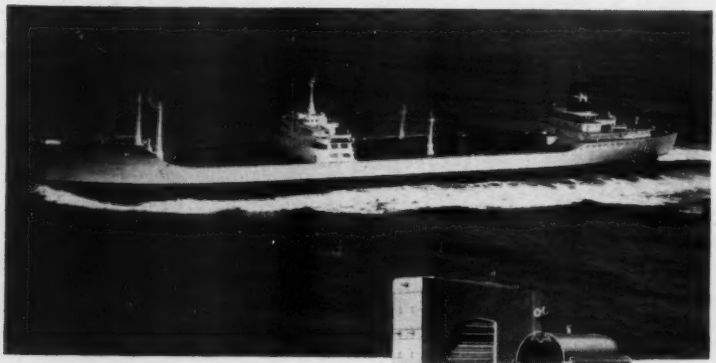

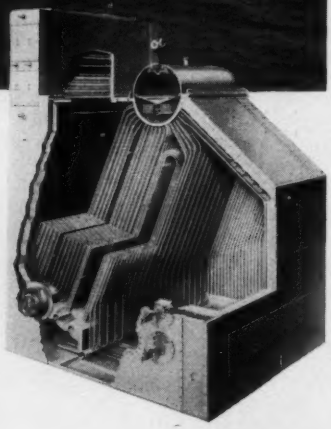

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Designed speed	18 knots
Number of cargo tanks	21

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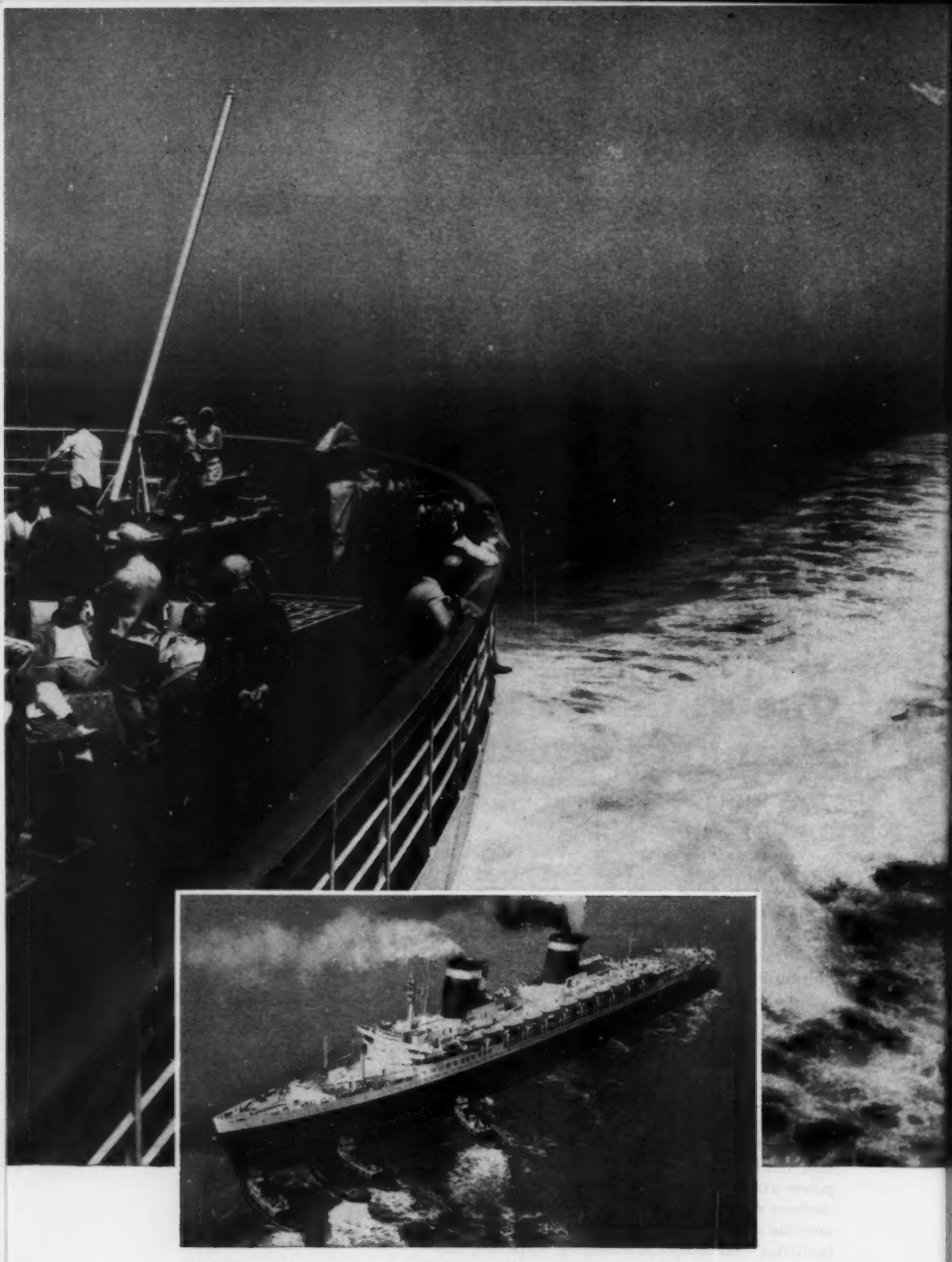

More than a quarter-million miles of water have passed under the stern of the S. S. United States since the vessel's maiden voyage July 3, 1952. While the actual horsepower data cannot be revealed for security reasons, during two successive days on the maiden voyage, she averaged over 36 knots or nearly 42 statute miles per hour.

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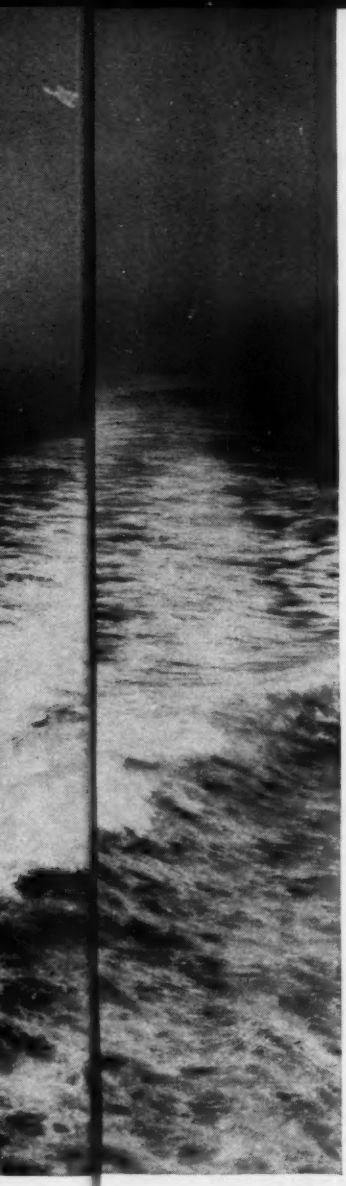

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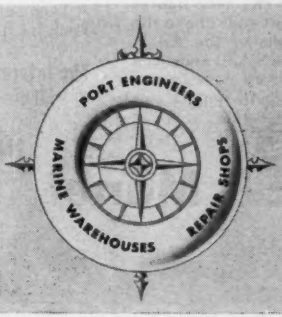

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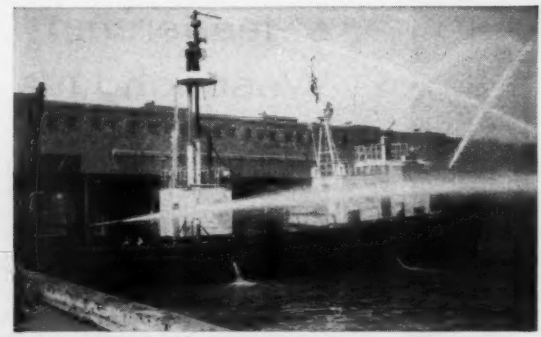


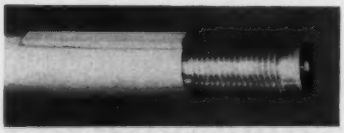
New fireboat, "Phoenix" — When her three main 550 hp Cummins diesels are pumping, two 110 hp diesels can drive her Monel shafts for maneuvering. The *Phoenix* is just over 88 feet long, has a 19-foot beam and a 6-foot draft.

Why John G. Alden chose
Monel shafts for new fireboat

Compared to San Francisco's two older fireboats, the new *Phoenix* is small and compact. But she's got plenty of pumping capacity, 9600 gallons a minute.

And she's fast. She can rush protection to the scene of any fire in San Francisco's great harbor at 15 miles per hour. That's almost twice the speed of the older fireboats.

Along with this pumping capacity and speed, dependability was a primary objective of the *Phoenix's* famous designer, John G. Alden, of Boston, Mass.



To meet requirements of the American Bureau of Shipping, this Monel propeller shaft had to pass the same tests a steel shaft meets. The test figures showed the Monel shaft well above the minimum requirements of the A.B.S. And note the use of the S.A.E. approved filleted type of keyway for greater strength and resistance to impact.

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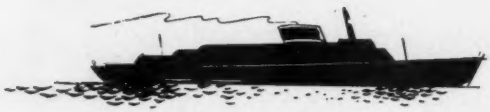

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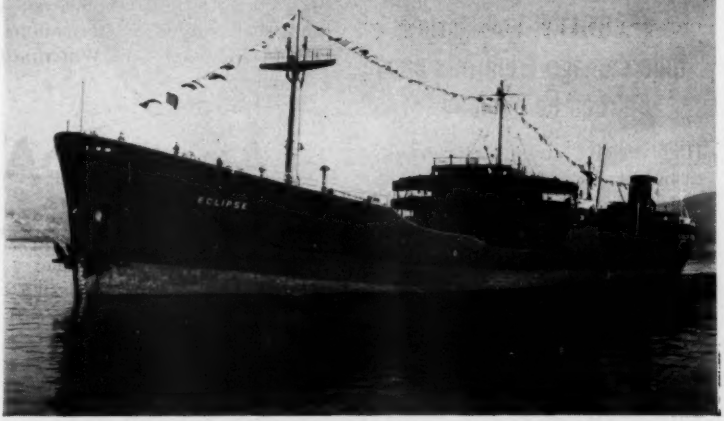

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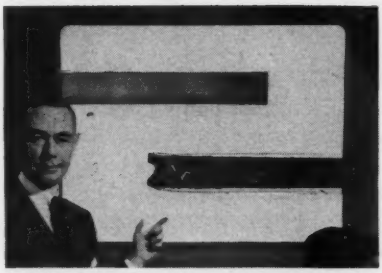

High-velocity, turbulent sea water damaged these Admiralty tubes. Top, flow over pebble or shell left small scar. Bottom, tube inlet is badly corroded and eroded.

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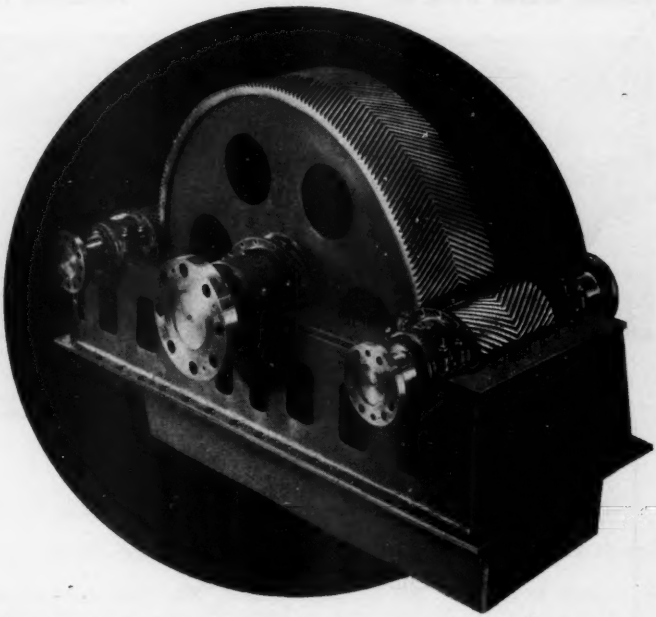

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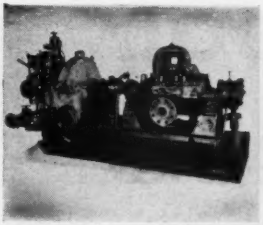
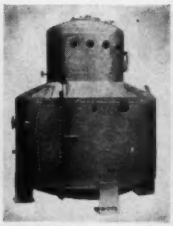
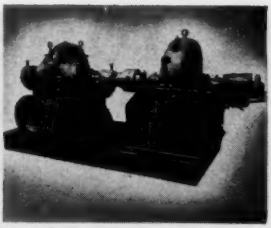

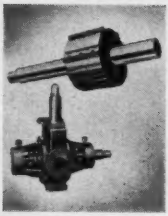
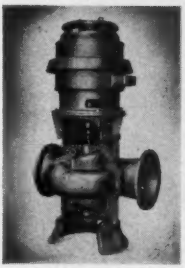
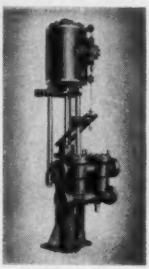
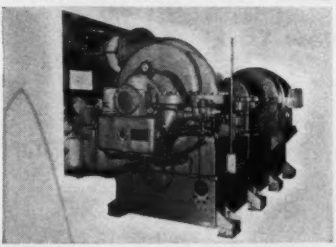
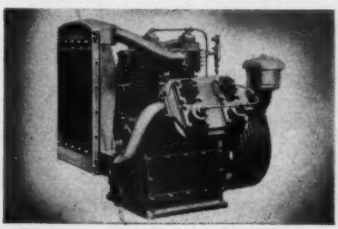

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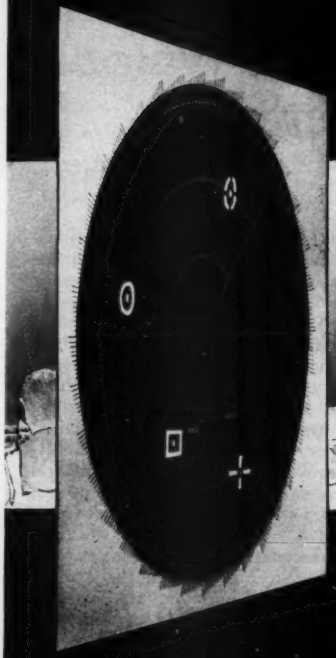

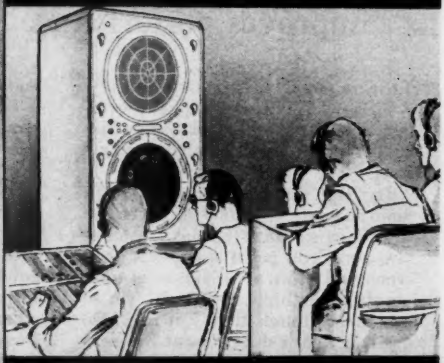

Developed by RCA in close coordination with Naval personnel, the Mark 6 Target Designation System is now in full production. It is but one of the many complete electronic systems which RCA has developed for the armed forces. RCA engineering — from original planning to final production — assures greater efficiency, effectiveness and safety in operation.

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IN this new engineering material—USS "T-1" Steel—you get a combination of mechanical properties never before obtainable in a single steel.

In "T-1" you get great strength (a yield strength of 90,000 psi.), yet you can fabricate this steel easily and at low cost. You get a steel with good creep and rupture resistance at temperatures as high as 900° F., yet so inherently tough that you can also use it in heavy duty jobs at *sub-zero* temperatures down to −40° F. In "T-1", in brief, you get a steel that withstands severe impact abrasion and, at the same time, resists corrosion at all temperatures.

This unique combination of properties helps you to cut costs and improve performance in an extremely wide range of industrial applications.

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UNITED STATES STEEL

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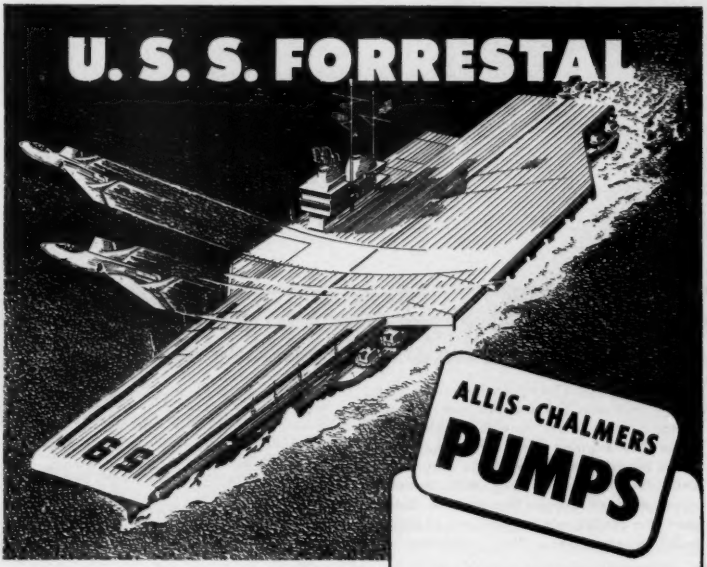

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9	6x6 SF	Salt water for gasoline system
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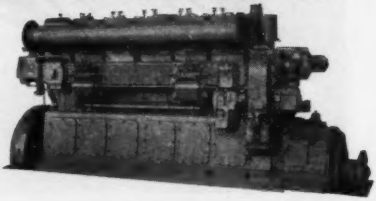

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Official United States Navy Photograph

CARRIER FORRESTAL CHRISTENED: Mrs. James Forrestal, widow of the first Secretary of Defense, christens the new aircraft carrier named in his honor at Newport News, Va. Also present were (left to right) Secretary of Defense Charles E. Wilson, Secretary of the Army Robert T. Stevens, Mrs. Byrnes MacDonald, Matron of Honor; Mr. Henry Forrestal, the late Secretary's brother; Mr. Peter Forrestal, the late Secretary's son; Secretary of the Navy Charles S. Thomas; Mr. William E. Blewett, Jr., President of the Newport News Shipbuilding and Drydock Co., and Mrs. Forrestal.

Official United States Navy Photograph

Worm's eye view of the Navy's newest carrier, the USS *Forrestal*, prior to the christening ceremonies.

Official United States Navy Photograph

Rear Admiral W. D. Leggett, Jr., U.S. Navy, Chief of the Bureau of Ships and 1955 President of the Society.

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JOURNAL

OF

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VOL. 67 FEBRUARY, 1955 No. 1

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(Under whose supervision this number is published)
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Captain Rawson Bennett, U.S.N.
Commander C. H. Campbell, U.S.N.R.
Rear Admiral R. E. Cronin, U.S.N.
Rear Admiral F. R. Furth, U.S.N.
Mr. John F. Hanlon
Mr. James S. Melton
Commander J. W. Naab, U.S.C.G.
Lieut. Comdr. J. W. Sawyer, U.S.N.R.
Captain Hugh Webster, U.S.N.

Captain J. E. Hamilton, U.S.N., (Ret.), Secretary-Treasurer

SECRETARY'S NOTES

Council Meeting

The annual meeting of the retiring and newly elected members of the Council was held on 5 January 1955. Matters which were considered at this meeting are presented below. See photograph on page 9.

Annual Election

The report of the ballot counting committee was received by the Council from its chairman, Captain T. W. Rogers, USN. Based on this report, the Council declared the results of the election to be:

Elected President for 1955:

Rear Admiral W. D. Leggett, Jr., U.S.N.

Elected Secretary-Treasurer for 1955:

Captain J. E. Hamilton, U.S.N., Ret.

Elected as Regular Naval Officer Members of the Council to serve in 1955 and 1956:

Captain Rawson Bennett, U.S.N., and
Rear Admiral R. E. Cronin, U.S.N.

Elected as Reserve Officer Member of the Council to serve in 1955 and 1956:

Lieutenant Commander J. W. Sawyer, U.S.N.R.

Elected as Civilian Member at the Council to serve in 1955 and 1956:

Mr. James S. Melton

The Council named Captain R. B. Madden, U.S.N. and Mr. A. G. Fessenden to continue during 1955 as Assistant Secretary-Treasurer and Administrative Assistant, respectively.

Resignation of Captain Heiser

When Captain H. M. Heiser, U.S.N., was detached from duty in Washington to become Supervisor of Shipbuilding at Pascagoula, Mississippi, he submitted his resignation as a member of the Council. To fill the vacancy thus created the Council appointed

Rear Admiral F. R. Furth, U.S.N.

to serve out the term which expires on 31 December 1955.

Junior Membership

The members were asked on the annual ballot to vote on a change of the by-laws as follows:

"Following article 18, insert:

"19. Undergraduates at schools to be named by and under regulations to be issued by the Council, shall be entitled to Junior Membership at one-half the dues of regular members during their course and for one year following graduation, after which time such junior membership will be terminated unless the junior member transfers to regular membership in the appropriate category."

The change was approved by a large majority.

As a result of the approval the Council took the following action:

a.) As an interim measure, the Secretary-Treasurer was directed to prepare a suitable application blank, and to send a number of these with a description of the Society and the new type of membership to the appropriate authority of each of the following schools with an invitation to each member of the next graduating class to take advantage of this opportunity:

U. S. Naval Academy
U. S. Military Academy
U. S. Coast Guard Academy
U. S. Merchant Marine Academy
The 52 Naval ROTC colleges for engineering students only.
Webb Institute

The four State Maritime Academies, namely:
California
Maine
Massachusetts
New York

This action was taken as an expedient so that the Class of 1955 at these schools would not be deprived of the opportunity to join the Society at the reduced rate.

b.) As a formal and permanent measure, subject always to modification by the Council, the President continued the committee which had originally studied the matter of Junior Membership with an additional member. This Committee as now constituted is:

Commander C. H. Campbell, U.S.N.R.
Mr. John F. Hanlon
Commander J. W. Naab, U.S.C.G. and
Lieutenant Commander J. W. Sawyer, U.S.N.R.

The Committee was assigned the mission of recommending a full list of schools to whose students this privilege should be extended and a proposed draft of the regulations to be issued by the Council to carry out the wishes of the Members in respect to Junior Memberships.

It is expected that Committee and Council action will be completed in time to take effective action before the opening of the 1955-6 academic year.

Honorary Membership

Article 19 of the by-laws provides for the election of Honorary Members. This honor has not been accorded to any member, at least within recent years.

One member has now been recommended to the Council as a candidate for Honorary Membership. The first step required has been taken. The Council has voted unanimously to present this name to the membership on the next regular election ballot. As an interim measure the Council has elected this member to Life Membership, which is to become effective at the annual banquet on 29 April 1955 at which time it will be announced.

This notice is being published, by direction of the Council, to call attention of all members to the by-laws' provision for Honorary Membership. It is certainly not intended to "push" this provision but in view of the action which has been initiated by one alert member, it is believed that other possibly eligible candidates for consideration for election to Honorary Membership should be brought to the attention of the Council.

No criteria for this nominationn have been established for the present. It is suggested that, if any member believes that there is some individual to whom the Society should extend this honor, the

name be submitted to the Secretary-Treasurer at any time. The nomination must be accompanied by a citation which will fully justify the recommendation. Any recommendation which is received will be considered by a recently named Awards Committee which will make its recommendation to the Council. It is necessary for the candidate to receive the unanimous approval of the Council and then, at the time of the annual election, to receive an affirmative vote by at least two-thirds of the members voting.

The action of the Council in votinng Life Membership to the one candidate is based on specific considerations and is not a necessary precedent to be followed for other nominations to Honorary Membership.

The Awards Committee which has been established consists of:

The Secretary-Treasurer, ex officio to act as chairman and three additional members who will serve at the pleasure of the President. Those who have been appointed are:

Captain W. A. Dolan, U.S.N.
Mr. J. J. Kennedy
Mr. John Neidermair

The Annual Banquet, 1955

Arrangements are being made for the 1955 Annual Banquet of the Society. It is to be held, as customary in Washington, D.C. The date is Friday, 29 April 1955 and, as in the case of all banquets since they were resumed after World War II, will be held at the Statler Hotel, 16th and K Streets, N.W.

Each member will receive a formal notice and application blank for the banquet during the early part of March. No application will be considered prior to this formal announcement and then only on the official form.

In general, the 1955 banquet will be arranged to the same standards as have been adopted for the last few.

The banquet will be presided over by the President of the Society, Rear Admiral W. D. Leggett, U.S.N. Admiral Leggett will also fill the role of toastmaster which previously has been a separately assigned function.

Vice Admiral Earle W. Mills, U. S. Navy, Retired

One of the Society's past Presidents, Vice Admiral Earle W. Mills, U.S. Navy, retired, has been elected to the Presidency of the Society of Naval Architects and Marine Engineers for a two year term starting 1 January 1955. Admiral Mills' last appointment while on active duty was as Chief of the Bureau of Ships and Engineer in Chief of the U.S. Navy, in which he served with distinction. At the present time he is President and Chairman of the Board of Foster Wheeler Corporation. He served as President of the Society in 1945.

Mr. Warren T. Johnson

The Society and its staff lost a valuable friend in the death on 16 December 1955 of

Mr. Warren T. Johnson

The exceptional interest which Mr. Johnson displayed in nthe welfare of the Society—its membership, its banquet, its meetings and its Journal led to the decision that a friendly tribute was preferable to a formal obituary.

His vital statistics and career steps have been stated elsewhere. His friends are much more interested in Mr. Johnson as we knew him in the last few years of his life.

Whenever the Society staff needed help in any of the places with which Mr. Johnson was familiar, and these embraced most of the Federal Government, he either spotted the need in advance and proffered assistance or, on request, went into immediate action. If any individual who was eligible to membership in the Society fell under his observation, an application blank was usually received by the Secretary-Treasurer. In his perusal of technical literature he was quick to spot and suggest articles worthy of reporting or books deserving of review. At the time of his death, Mr. Johnson was serving on the Society's Ballot Counting Committee and his name was included among those who were to serve on the 1955 Banquet Committee. He never said "no" to any demand placed on his time or energy, by a member of the Society staff.

We began with the statement that the Society had lost a friend. There appears no better way to end that to repeat this—a long-to-be remembered and missed friend.

MISCELLANEOUS

Joint Engineering Societies Meeting

For sometime the Engineering Council in Washington, D.C. has been working on plans to hold a joint meeting of all Engineering Societies in the nation's capital. It was and is hoped that this meeting would lead to others until an annual meeting, representative of all engineers, would be a regular and popular Washington affair.

The Society is a member of the Council and had expected to announce the meeting in the Journal. Unfortunately decisions as to the time and nature of the meeting had not been reached when the last issue went to press. The date now selected is 23 February 1955, two days before this issue appears. We intend to report on this meeting in the May issue and hope to be in a position, for future meetings to contribute to their success in any way which is available to us.

American Society of Safety Engineers

The Society has been requested to establish cooperative relations with the American Society of Safety Engineers. Naturally we are in full accord with the aims of that Society and have agreed to cooperate in any way which presents itself.

It is our feeling that most members are brought into touch with the activities of the National Safety Council with which the ASSE is affiliated during their normal pursuits and thus, as far as aims are concerned, our members are "cooperating" in a practical way. Their Society will represent them in cooperation as opportunity arises.

U. S. Government Research Reports

The Office of Technical Services of the Department of Commerce is urging that members of engineering societies, groups and associations make greater use of research reports which are released through that office.

Resulting from the Federal Government expenditure, in sponsorship of scientific and technical research, which at the current rate is $2 billion per year, are some 350 such reports each month. These are described in a monthly publication "U.S. Government Research Reports" which is available from the Superintendent of Documents, U.S. Government Printing Office, Washington 25, D.C. or any of the U.S. Department of Commerce Field Offices at $6 a year.

Erratum

It is regrettable that Fig. 1 of Dr. Melvin Mark's article, "Aerodynamic Torque Loads on Rotating Radar Reflectors," was inverted and printed upside down. This is on page 953 of the November 1954 issue.

tain H. M. Heiser, U.S.N., Council 1954.

Chase Ltd., Photo, Washington, D.C.

JOINT COUNCIL MEETING—5 JANUARY 1954. Left to right, Front Row: Mr. A. G. Fessenden, Administrative Assistant; Mr. J. S. Melton, Council 1955-6; Rear Admiral R. E. Cronin, U.S.N., Council 1955-6; Rear Admiral W.D.Leggett,Jr., U.S.N., President 1955; Rear Admiral E. W. Sylvester, U.S.N., President 1954; Lieutenant Commander J. W. Sawyer, U.S.N.R., Council 1955-6; Captain Rawson Bennett, U.S.N., Council 1955-6. Back Row: Mr. John F. Hanlon, Council 1954-5; Captain Hugh Webster, U.S.N., Council 1954-5; Commander J. W. Naab, U.S.C.G., Council 1954-5; Mr. H. E. Carleton, Council 1953-4; Captain T. H. White, U.S.N., Council 1954; Captain R. E. W. Harrison, U.S.N.R., Council 1953-4; Commander C. H. Campbell, U.S.N.R., Council 1954-5; Captain J. E. Hamilton, U.S.N., Ret., Secretary-Treasurer; Captain R. B. Madden, U.S.N., Assistant Secretary-Treasurer. Absent: Rear Admiral F. R. Furth, U.S.N., Council 1954; Captain H. M. Heiser, U.S.N., Council 1954.

Official United States Navy Photograph

The USS *Forrestal* being moved into pier eight, Newport News, Va., after christening ceremonies.

HALF A CENTURY OF PROGRESS IN NAVAL BOILER WATER TREATMENT

FRANK E. CLARKE

THE AUTHOR

graduated from Western Maryland College in 1935 with degrees in chemistry and education. Studied advanced chemistry for one year in University of Maryland and taught chemistry for five years before entering the Water Section at the EES Chemical Laboratory in March 1941. He became head of the Water Section in 1942, Head of the Chemical and Physical Branch in 1946 and Head of the Process Branch in 1951. He has handled water treatment and water side problems for the Fleet since 1942.

The importance of boiler water quality has increased steadily since the Fleet launched its first steamship nearly a century and a half ago. (Originally USS *Fulton*, 1814, later USS *Demologos*). Naval personnel have been exceptionally self-sufficient in solving their boiler water problems and have pioneered a creditable portion of boiler water science. This does not mean that all problems have been solved expeditiously or completely. Steady advances in boiler design have taxed the ingenuity of water technologists. Correction of obvious disorders frequently has disclosed more remote and stubborn troubles.

During its half-century of existence, the Engineering Experiment Station has developed Naval water treatment from an art of the 19th Century, frequently the black variety, to a matured science characteristic of the 20th century. The present trend in Naval propulsion indicates that this will be a busy science for many years to come.

BEFORE THE CENTURY TURNED

The 19th Century record of Naval water treatment is a shabby volume with many obscurities and missing pages. However, it shows that chemical boiler water treatments already were in effect. Lt. Comdr. Frank Lyon (Figure 1) in recording his experiences as assistant engineer on the USS *Oregon* (1896 to 1899) stated "the feedwater was kept hot in practically an open tank and was at all times kept so strongly alkaline as to render it unfit for ordinary purposes. In this vessel corrosion of the boilers or of the piping

was almost unknown." The alkalinity probably was obtained with either soda ash or lime since both were in use at that time. There is evidence that numerous other materials were used for water treatment before 1900, including legendary treatments like potato rinds. The promoters of magic cure-all devices also were vending their wares.

Generally speaking, 19th century Naval boiler water treatment was aimed at preventing corrosion rather than scale and sludge deposition. Waterside scale frequently was developed with lime to reduce corrosion. There was decided lack of standard treatment and testing programs. Individual engineers were permitted considerable latitude in using proprietary treatments and pet schemes. Frequent mechanical cleaning and the inherent ruggedness of Scotch boilers were important factors in keeping the Fleet underway.

Figure 1. Frank Lyon, Father of Naval Water Treatment (1910).

THE FIRST DECADE OF A NEW ERA (1900 TO 1910)

Waterside problems mushroomed during the first several years of the new century primarily because watertube boilers were introduced in appreciable numbers. Frank Lyon, now senior engineer of the USS *New Jersey* (1906-1909) reported serious trouble from corrosion in the vessel's Babcock and Wilcox "smalldrum" boilers (presumably watertube). These boilers were kept only slightly alkaline, first with soda ash and later with lime. Higher alkalinities caused carryover. Lime was substituted for soda ash because it "floated the oil to the surface better." Installation of zincs did not eliminate the corrosion problem. Lyon left this billet for a three year tour at the infant Engineering Experiment Station (EES) knowing there was something radically wrong with Navy regulations for controlling boiler corrosion.

During this same period other Naval officers were attempting to improve boiler water treatment. An undated memorandum from Lt. H. C. Dinger

(Captain Dinger directed EES 1926-1930) to the Bureau of Engineering suggested commercial boiler compounds for the "conditions generally met with in Naval boilers namely, oil and dirt deposits with slight salt scale." The memorandum mentioned sal soda, (hydrated sodium carbonate), caustic soda, kerosene and turpentine, then used to remove oil and dirt, and recommended that the matter of scientific boiler compound be taken up with Dearborn Day and Chemical Company, Chicago; Coralline Drug and Chemical Company, Chicago; and Reid-Archer Company, New York. In another communication dated 1907 the Chief of the Bureau disapproved the use of lime in boilers and warned against the use of excessive soda ash for oil control. The communication closed with the statement "to minimize the corrosive effect of air, the Navy regulations prescribe the best means known at present."

By the end of the decade, a considerable number of water tube boilers had been in service for periods ranging from 4 to 7 years and their peculiarities were becoming apparent. In 1909 Rear Admiral George W. Melville, founder of EES, reported that water tube Thornycroft boilers installed on the USS *Missouri* in 1903 were troubled with waterside grease, salt and dirt. He said a great deal could be done with soda and other cleansers as well as with frequent blowdown, examination and mechanical cleaning. This ASNE note also mentioned priming troubles from condenser leaks, the relatively low tolerance of water tubes for scale and the impracticability of removing heavy waterside scales by mechanical means. In a similar ASNE note in 1910, Rear Admiral R. H. S. Bacon quoted from "Power and the Engineer" a long list of water contaminants, their individual effect and remedial measures for the principal ones in Naval service.

TABLE 1—RECOMMENDED MARINE BOILER TREATMENT—1910

Water Contaminant	*Remedial Action*
Mud	Blowdown
Lime bicarbonate and magnesia	Add caustic soda, lime or magnesia
Sulfate	Add soda carbonate or barium chloride (not too much or foam will result)
Magnesium sulfate	Add soda ash
Acid	Add alkali

The year 1909 marked the beginning of Frank Lyon's classic experiments in boiler water treatment at EES that placed the Navy in water treatment's hall of fame. While these experiments were not completed until the beginning of the second decade, the work finished in 1909 and 1910 left little doubt that Navy water technology had put on its long pants.

THE SECOND DECADE—A LYON'S SHARE

Frank Lyon was not an expert water technologist when he started his investigation of boiler corrosion at EES in 1909. In witnessing a surgical operation about that time he was amazed to find the nurse storing instruments in a basin of water containing several spoonfuls of sodium carbonate. Lyon admitted "that was the writer's first intimation that there was anything

with water in it that was less corrosive than good neutral distilled water." Despite this meager knowledge he started his three years of research like a seasoned investigator and compiled an amazing amount of useful data.

Electrolytic Studies

In view of experiences on the USS *New Jersey*, it is not surprising that he concentrated primarily on understanding and preventing corrosion of steel and non-ferrous shipboard alloys. Work started with measurement of electrical potential differences in aqueous solutions between metal pairs, stressed and normal metals, and corrosion products and parent metals. In some tests potentials were measured instrumentally. Others used the ferroxyl mount technique of imbedding specimens in gelatin containing potassium ferricyanide and phenolphthalein. The corrosion areas (high potentials) and passive areas develop blue ferroferricyanide and pink phenolphthalein colors respectively. These still are favorite techniques for predicting corrosion characteristics. From these studies, Lyon learned that every metal specimen contains a multitude of positive and negative areas that keep changing locations in corrosive media. He found that polarities of identical specimens keep switching so that they share the positive position and weight loss. Stressed areas like gouged or poorly rolled tube seats proved to be relatively corrodible, while corrosion products were negative to their metals. An interesting aspect of the latter was discovery that old zinc can actually cause corrosion of steel, though new zinc protects it.

Corrosion Tests

The potentiometric studies were followed by static corrosion tests in jars of distilled water, sea water and Severn River water. Specimens consisted of steels and irons immersed separately or in contact with non-ferrous alloys including monel, Muntz metal, bronze, copper and brass. Some of the jars were left sealed for periods up to two years after which they were opened and evaporated to dryness. Others were left open and fortified regularly with water to make up evaporation losses. In one or two special instances, oxygen and air were bubbled in the solutions to determine their effects. These tests showed that all metals corroded to some extent in all waters, that iron corroded less in sealed jars than in open jars and that the effects of bimetallic couples depended to a large extent on the degree of contact between the two. They also demonstrated that graphite in contact with steel caused it to corrode and that suspended mineral oil would not prevent corrosion.

The high corrosion rate in open jars was attributed to removal of corrosion products by oxidation and consequent reduction in solution potential. This

Figure 2. First EES Experimental Boiler (200 psi) Captured From Spanish Launch 1899.

TABLE 2—CORROSION DATA, OPEN JARS, DISTILLED WATER (LYON 1910)

Specimen	Weight Loss, Milligrams Per Square Inch		
	First 30 Days	First Year	Second Year
Polished Steel Boiler Plate	12.1	186	158.5
Same in contact With Copper	17.9	194.4	192.0
Same with Mill Scale except on Edges	12.1	207	182.4

was not a bad guess for 1911. It agrees with the modern theory that oxygen combines with corrosion products rather than directly with the metal. It differs with the present concept that the principal effect is removing atomic hydrogen from the cathode area, but the basic idea was similar. The observation regarding negative effects of suspended oil also was far ahead of its day. Most early boiler operators were convinced that a little oil was a good thing. Table 2 shows typical data collected during the jar tests.

Treatment Studies

Lyon's final studies, and the ones for which he is most famous, consisted of determining the effects of various chemicals on corrosion rates of steel. These tests were made with distilled water and various concentrations of sea water and Severn River water, using opened jars, closed jars, steel pots, and the experimental boiler shown in Figure 2. This was the first Navy experimental boiler used for water treatment studies. Treatment chemicals included sodium carbonate, caustic soda, disodium phosphate, sodium acetate, sodium and potassium chromate and dichromate, sodium silicate, borax and lime. This impressive list includes caustic soda and disodium phosphate, the most universally favored team of water treating chemicals today, and chromates which still are the best corrosion inhibitors for saline solutions.

TABLE 3—PERTINENT CHEMICAL CONCENTRATIONS, BOILER STEEL, DISTILLED WATER (LYON)

Treating Chemical	Concentration of Treating Chemical % Normal		
	Upper Limit	Critical Limit	Lower Limit
Sodium carbonate	2.6	0.8	0.16
Caustic soda	2.6	0.8	0.16
Lime	2.6	0.6	0.15
Disodium phosphate	4.0	1.2	0.16
Sodium chromate	0.26	0.08	0.03
Potassium chromate	0.26	0.08	0.03
Sodium dichromate	0.26	0.08	0.03
Potassium dichromate	0.26	0.08	0.03

The test solutions were prepared in various chemical normalities. This was a marked improvement over the standard practice of using chemicals by weight percentages, regardless of their chemical equivalent weights. They were analyzed for active ingredients instead of assuming correct concentration on the basis of weight data. Metal losses were plotted against concentrations of active ingredients. The curves showed that general corrosion occurred immediately on all specimens in distilled water. Increase in concentrations of certain treating chemicals appeared first to decrease corrosion, then to increase it slightly and finally to cause a sharp decrease to essentially zero. The initial increase in corrosion was said to consist of localized pitting.

From these data Lyon selected three limits: lower, the concentration just below first appearance of pitting; critical, the point of maximum pitting, and upper, the concentration just above that which prevented corrosion entirely. Table 3 shows values for these points on several of the chemical solutions tested.

A typical graph used by Lyon in selecting pertinent chemical concentrations is given in Figure 3. It shows that the length of exposure had very little effect on the shape of the corrosion curve. These curves do not appear to agree with the above table for the same conditions. For example the graph shows maximum Corrosion (critical limit) at 0.1% normal (line C^3) though the table gives it as 0.8% normal, or just to the right of line C^2. The fact that points 0.8 and 2.6 do not occur on the graph could mean that these pertinent concentrations were not included in the plots. At any rate 2.6% normal alkali was selected as the upper limit for iron and steel. This meant that 2.5% normal was the highest concentration in which corrosion had occurred. This was the origin of the Navy expression percent normal alkalinity which endured for almost 40 years (to 1946).

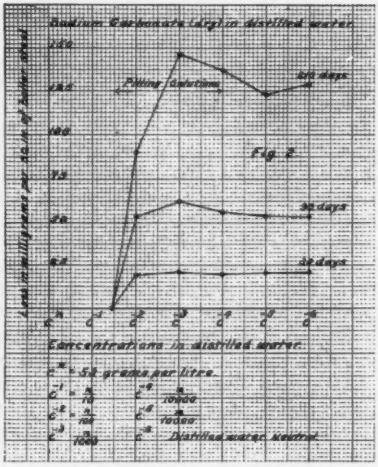


Figure 3. Effect of Alkali on Corrosion (Lyon, 1911).

Boiler Compounds—On the basis of the above studies, Lyon concluded "any boiler using any water can be kept from corroding for any length of time if treated with soda and if its concentration is maintained at or above 3% normal alkaline strength. If the water is not to be kept sufficiently alkaline, it had better be kept neutral." He started searching for boiler compounds to meet the requirements below:

(a) Render the water non-corrosive at any temperature apt to be encountered.

(b) Hold in suspension or preserve in colloidal form all of the salt in the water that otherwise would precipitate as scale, whether such precipitation be by the action of heat or by that of any chemicals used in the compound.

(c) Render the treated water no more liable to foam or prime than distilled water would be under the same conditions.

(d) Be composed of chemicals that are safe to store, that do not deteriorate materially with age and that are non-explosive and non-inflammable.

(e) Contain a constituent whose strength in the water (concentration) is easily measured by some simple chemical measuring device.

His quest included both proprietary and non-proprietary mixtures. His observation below concerning the former is amusing and as appropriate today as it was then.

There are now on the market many chemical compounds for treating boiler water. The claims made for these compounds are many and varied. Listening to the agents for almost any one of them, one would be tempted to believe that it would do anything in the cleaning and scale removing line, even to the absurd and the impossible. One agent claimed that his compound has the peculiarity of changing the form of the scale forming molecules in the water from the hook shape to the spherical. These molecules, being round, cannot stick on the metallic surfaces, and therefore cannot form scales. The writer unfortunately cannot verify this claim.

Every compound was tested for chemical composition, corrosion prevention, scale removal, scale prevention and priming characteristics. Analysis by the Station's chemist included qualitative and quantitative determinations for principal ingredients, specific gravity checks on concentrations 1/10th, 1/100th, and 1/1000th the original (liquid compound) and a study of storage hazards and deterioration tendencies.

Corrosion tests were made in jars and the experimental boiler. Liquid solutions were used full strength and in many dilutions. Powders were dissolved in distilled water (50g/liter) and then treated as liquids. Distilled water and Severn River water were used in preparing the final test media. The original report gave Figure 4 as a typical corrosion curve for most compounds studied.

Scale removing properties were determined by dissolving the compound in Severn River water, evaporating the mixture to dryness in an iron tube, baking it for one hour, refilling it with the same water, and finally inspecting for removal of the baked deposit. Practical tests for prevention and removal of scale were made on deposits developed by evaporating Severn River water in the experimental boiler until the chloride reached 3,000 grains per gallon. A part of the deposit was removed from the crown sheet, the boiler was resteamed with treated water and scale measurements were made on the test spot and adjacent areas. Antifoam tests were made by adding boiler compound solutions to laboratory distilling flasks of boiling caustic soda solution.

Some of these early compounds were dangerous. The report mentioned one composed of hydrochloric acid and a graphite compound which removed scale simply by causing electrolytic corrosion of the metal. Lyon reported that some compounds were good when used properly, that some were bad under any circumstances, and that all were bad when used improperly. His advice to the Fleet was, "unless a compound is used properly it is better let alone."

The fruits of the three-year investigation were the Navy's first standard boiler compound, boiler water testing cabinet, and instructions for boiler water testing. Why commercial compounds were rejected in favor of the non-proprietary mixtures shown in Table 4 and what prompted selection of these ingredients and proportions is not apparent from the record.

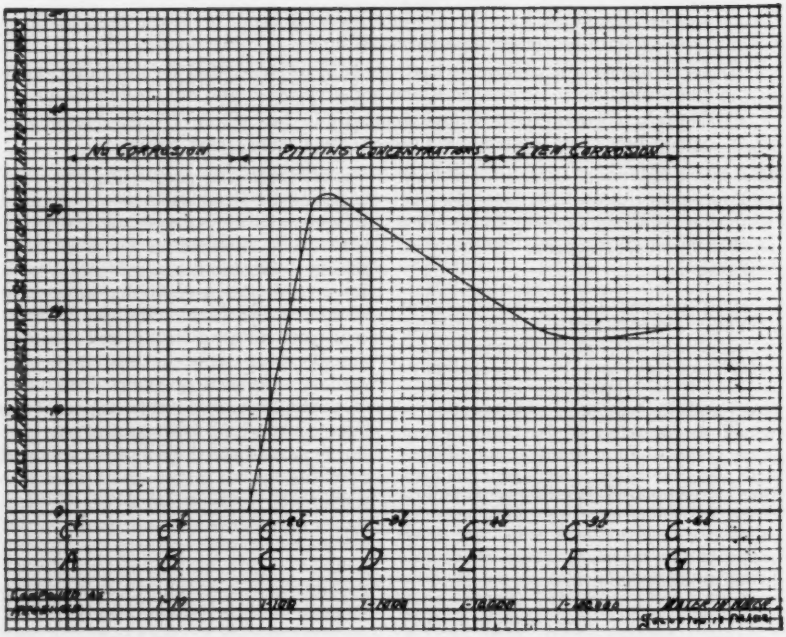


Figure 4. Effect of Commercial Boiler Compound on Corrosion (Lyon, 1911).

TABLE 4—FIRST NAVY BOILER COMPOUNDS (1911)

Constituent	*Lyon's Minimum Percent*	*First Specification 1BC3, Percent*
Sodium carbonate, Na_2CO_3	64	68 to 70
Trisodium phosphate, $Na_3PO_4.12H_2O$	15	19 to 22
Dextrin or starch	1	1 to 2
Tannin (35% tannic acid)	10	4 (Min)

Table 5 shows the purposes of the ingredients.

In 1913, the formula was changed as shown in Table 6 for some unrecorded reason. This was the last change for twenty years.

The first testing cabinet shown in Figure 5 contained glassware and reagents for alkalinity and chloride tests. In many respects it resembles the test cabinet still in use. The seven-page instruction which accompanied it was complete with calculations and examples. This combination of standard compound, test equipment and instructions puts the Navy's water treatment on a scientific basis for the first time. The whole subject was discussed in the 1912 edition of Marine and Naval Boilers by Lyon and Hinds.

Tradition indicates, although the record does not prove it, that Frank Lyon's prescription of 2.6% minimum alkalinity met with failure because of

TABLE 5
EXPECTED ACTION OF 1911 BOILER COMPOUND

Chemical	*Purpose*
Sodium carbonate	Prevent corrosion
Sodium phosphate	Reduce priming
Dextrin and tannin	Induce sludge instead of scale

TABLE 6
NAVY BOILER COMPOUND (1913 SPECN 13C3a)

Constituent	*Percent*
Soda ash	76
Trisodium phosphate	10
Tannin	2
Starch	1
Moisture	11

foaming troubles, and that he dropped back to a lower concentration level and settled for less than complete corrosion protection. An alkalinity value of 0.5% normal was prescribed in later editions of his boiler book.

Contemporary Investigators

While Frank Lyon held the lime light during this decade, Ensign E. G. Wilson and Comdr. C. A. Bonvillian made significant efforts to improve Navy boiler water treatment. The first concentrated on control of chloride contamination. He emphasized the sources of sea water leakage, recommended maximum limits of 50 grains per gallon for large tubes and 15 grains per gallon for small tubes and stated that every effort should be made to keep the salt concentration at much lower levels. This was the first mention of the chloride limits which still are in effect, though expressed in epm. A turbidometric test kit for fresh water chloride also was described. This used 10% silver nitrate, a much stronger solution than recommended by Lyon for boiler water testing and

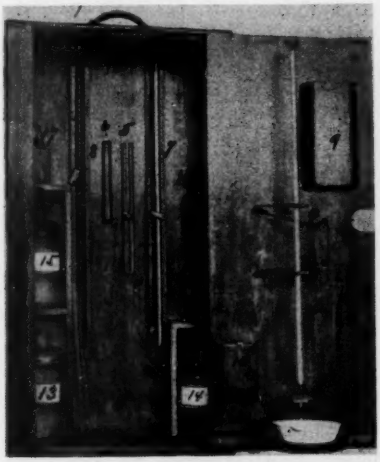


Figure 5. First Navy Boiler Water Test Cabinet (1912)—0.1N H_2SO_4, 0.024N $AgNO_3$, Potassium Chromate and Mixed Methyl Orange-Phenolphthalein Indicator.

constituted a second standard water testing kit. The same test still is used by submarines.

Commander Bonvillian was interested in improving instructions, supply, and personnel indoctrination rather than test methods. He pointed out the hazards of using shore water and stated that most evaporator plants were capable of yielding better water than the crews obtained from them. He criticized the freedom of judgment in applying shipboard treatments and tests as well as the abuse of Naval boilers in builders' yards. Most important, he recommended a system for inspecting, supervising, and troubleshooting shipboard water treatment. This was to be organized under the Fleet Engineer and consist of a board of officers from local ships. A standard boiler record sheet was designed for this program. There is no indication that the plan was approved.

NO ROARING IN THE TWENTIES

This period generally known for its glamour, prosperity and industrial progress was relatively quiet in the field of Navy water treatment. In a sense, Navy water technologists were resting on the laurels of Frank Lyon. The 1913 boiler compound still was in use for corrosion control and water side scale still was tolerated as a necessary evil. Some time during the decade, the limits for alkalinity were spread to the range 0.1 to 0.5% normal but this was a step in the wrong direction. A change from sulfuric acid to 0.5N nitric acid for the alkalinity test also seems to have occurred during this period, but the same testing cabinet was used. A third edition of Lyon's boiler book repeated his treatment theories in skeleton form and described a boiler compound, Table 7, not mentioned in the original report, and apparently never used by the Fleet.

TABLE 7
UNKNOWN BOILER COMPOUND—1920 EDITION "MARINE AND NAVAL BOILERS"

Component	*Percent*
Sodium carbonate	95
Disodium Phosphate, $Na_2HPO_4.12H_2O$	4
Cutch	1

A comprehensive 14-page note on standard navy boiler compound (Table 6) published by the Experiment Station in 1927 (ASNE) unwittingly admitted that the "experts" were confused about the principles of internal boiler water treatment. In one breath, it assures that navy boiler feedwater is "soft" and non-scaleforming and in the next it discusses the advantages of laying down a protective scale with milk of lime treatment. Poor advice was issued against excessive blowdown and unnecessary mechanical cleaning and supplementary treatments, like bichromate of soda were prescribed for "active corrosion." The inconsistencies, uncertainties and ambiguities show that the water technologist of the 20's was seeing the subject "through a glass darkly." It must have been disconcerting for the operator to receive the list of supplementary chemicals in Table 8 with the statement "there are numerous other materials that may be used for feed-water treatment and some of these, on occasion may be used in connection with navy boiler compound."

TABLE 8—SUPPLEMENTARY WATER TREATING CHEMICALS (1927)

Silicate of soda (Waterglass)	Lime
Sodium bichromate	Sal soda (hydrated sodium carbonate)
Caustic soda	Cutch
Kerosene	Flax seed
Miscellaneous materials to secure colloidal effects	

Conclusions like "a certain line of ships in certain service might find a certain variation from standard practice desirable" also must have left the operator completely confused.

The quiet 20s represented a darkness before the dawn of really scientific boiler water treatment. Numerous articles were published on express boilers with operating pressures up to 2000 psi. The disastrous effects of waterside deposits and dissolved oxygen pitting on these newcomers already were becoming apparent. The Engineering Experiment Station note referred to above closed with the statement "air-free steam is probably the

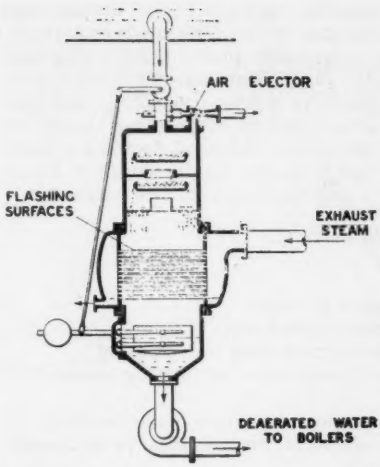


Figure 6. Early Flash Type Deaerator (1923).

next big thing to go after as far as preventing the deterioration of boilers, piping and turbines." Investigators were searching for something better than iron laths and turnings for oxygen removal. The 1923 issue of the ASNE journal published a note on mechanical deaeration from "Shipbuilding and Shipping Record" including the sketch of a flash deaerator shown in Figure 6. This was very similar to units tested and discarded by the Navy many years later (1941). Electrical salinity indicators also were being proposed (1927).

In another branch of the service (Bureau of Mines) Dr. R. E. Hall was busy explaining the real functions of soda ash and sodium phosphate in boiler water conditioning (Patented 1925 to 1927). Everett Partridge was conducting his classic investigation of boiler scales at the University of Michigan under the sponsorship of the Detroit Edison Company. These two investigators apparently inspired the Navy to take a new look at its boiler water treatment. In March 1929 Cdr. R. W. Payne of the Bureau of Engineering wrote to Capt. H. C. Dinger, Director of the Experiment Station, regarding the promising aspects of the Hall scheme of water treatment. He recommended an approval test of it at the Station and a possible service trial on the USS *Saratoga*. Dr. Hall apparently had an unusual reputation, for Captain Dinger replied "I see no particular thing to be gained by having the Experiment Station try to analyze Dr. Hall's methods. They have been sufficiently established and tried out in practice to know that if properly carried out they will produce satisfactory results." He recommended that the system be tried on the *Saratoga* in lieu of a Laboratory evaluation and suggested the possibility of training Naval personnel in the Hall Laboratory. His letter closed with a postscript request that the project be charged against "engineering" because the Station's appropriation was obligated to the full extent. This was a strange commentary for the golden age before the crash.

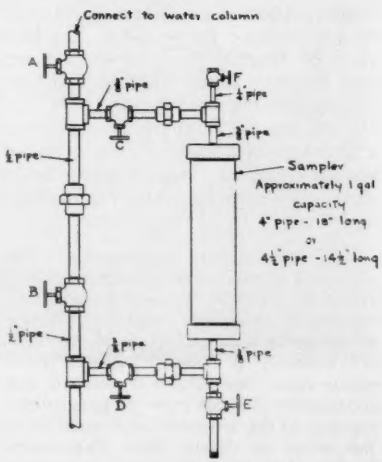


Figure 7. Nalco Air Cooled Boiler Water Sampler (1930).

The Hall test kit never got aboard the *Saratoga*. However, the desire for a better Navy water treatment persisted. In July, 1930, Capt. G. B. Vroom, the new Director of the Station, responded to "verbal instructions from the Bureau to consider methods of investigating feedwater treatment with particular reference to several commercial methods." The Station staff decided to start by analyzing representative samples of boiler water from the Fleet. They recognized the importance of uniform sampling and proposed that all cooperating vessels use the Nalco (National Aluminate Company) Sample Cooler shown in Figure 7 and the questionnaire below.

(a) Are there any troubles from scale formations?
(b) Any corrosion troubles?
(c) What is the blowdown practice?
(d) Are the reserve tanks coated with cement wash?
(e) Is there any other coating used in reserve feed tanks?
(f) Is the rate of scale deposit and corrosion increasing or decreasing?
(g) What has been the general practice in regard to maintaining alkalinity in the boiler water?
(h) What is the actual weight of boiler compound used per boiler per day?
(i) Describe how boiler compound is placed in feed systems, as to amount, method of introduction and frequency.

NAVY BOILER COMPOUND 1933— SOLBERG AND ADAMS

The decision of the Bureau of Engineering and the Experiment Station to investigate Naval water treatment on a grand scale began to bear fruit in the first year of the third decade. The problem was assigned to Lt. Cdr. T. A. Solberg (now Rear Admiral, retired) an experienced investigator. The Bureau of Engineering provided direct and indirect funds totaling approximately $30,000, a fortune for 1930. The United States Shipping Board (later dissolved) provided for salaries of special civilian technologists in a dramatic resolution which read in part as follows:

Whereas, the Navy Department of Engineering is engaged in a continuing project at the Navy Experiment Station, Annapolis, Maryland, to cover an extensive investigation into and study of the modern methods of boiler feedwater treatment—Now, therefore be it resolved, that the transfer of $7,500 from the unexpended balance of the aforesaid appropriation to the officer in charge Navy Experiment Station, Annapolis, Maryland, to be used in expediting and extending its investigation—.

Commander Solberg contacted Dr. White of the University of Michigan and requested the assistance of Dr. Partridge in planning the Navy's project (arranged later by Bureau of Mines). As a result of this conference, R. C. Adams, a fellow worker and co-author with Dr. Partridge, was hired to assist with the work in August, 1931. A sub-professional technician, originally Paul A. Parent and later Edwin L. Ford completed the team.

The new investigation started with simultaneous searching of the literature, solicitation of water and scale samples, from naval vessels, bench tests of corrosion and scale formation, and construction of experimental boiler equipment for extended studies. All told, 63 vessels cooperated in submitting samples or trying treatments. The seven types of samples requested are listed below.

(a) Boiler water after 300 hours of operation
(b) Boiler water just prior to boiler cleaning

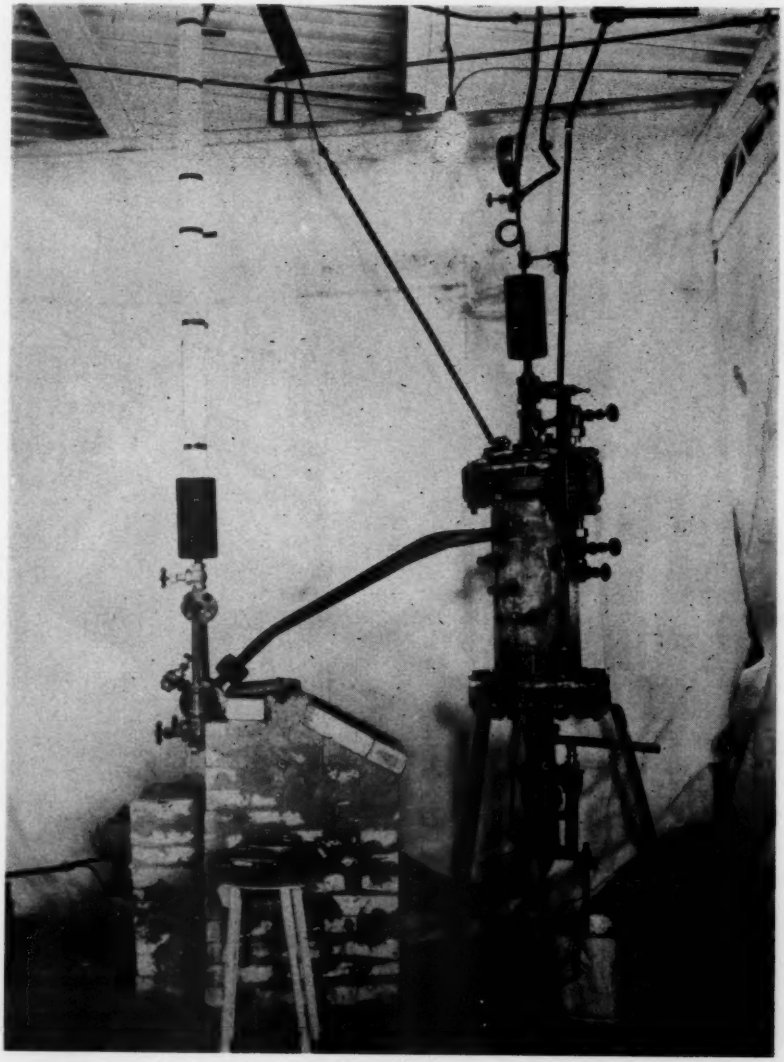


Figure 9. EES 100 psi Boiler X (1930).

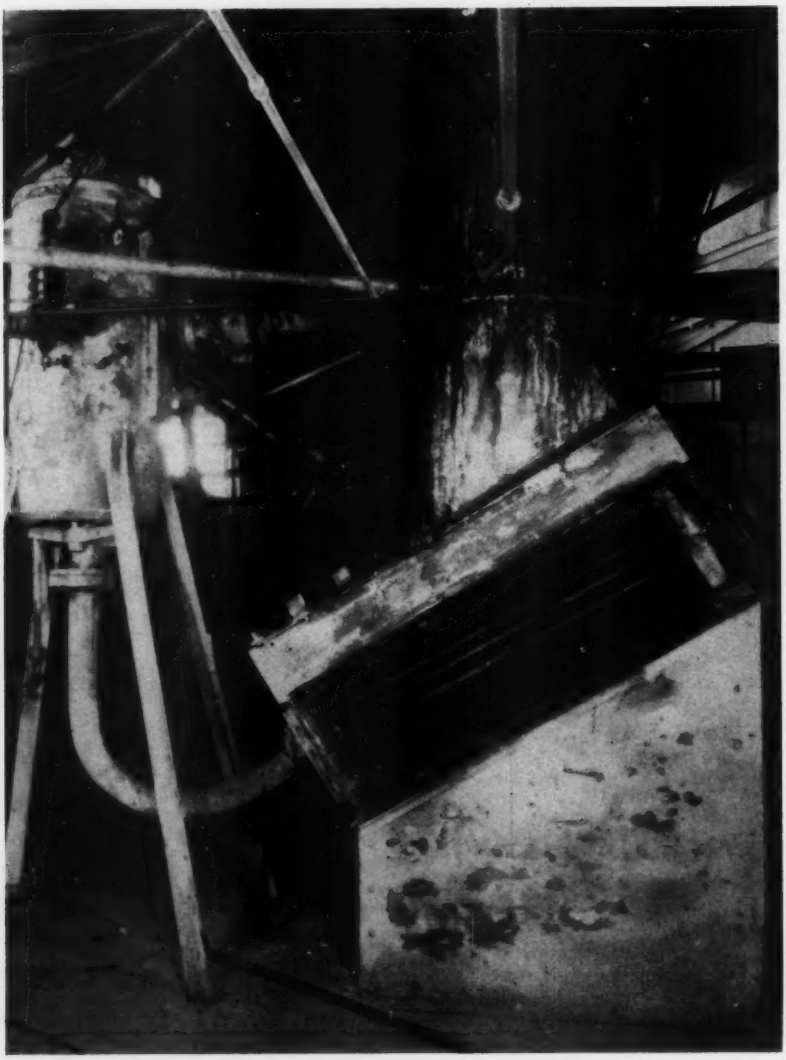


Figure 10. EES 225 psi Boiler A (1931).

(c) Reserve feed tank water
(d) Distiller discharge
(e) Air pump or condensate discharge
(f) Main pump suction line
(g) Overboard sample from open sea

The bench test included beaker corrosion tests, much like those of Frank Lyon, and scaling tests with immersion heaters and the autoclave in Figure 8. They were not particularly valuable because of lack of control and divergence from actual service conditions. Experimental boiler tests for corrosion and scale formation followed the bench tests. The first in Boiler X (Figure 9) were to determine dissolved oxygen effects. This three-tube, 100 psi boiler was constructed from scrap prior to the arrival of Solberg and Adams. The tests confirmed the corrosive nature of oxygen but showed that it had very little influence on scale formation. Experimental boiler tests of various proprietary and laboratory boiler compounds continued for approximately two years using the boilers shown in Figures 10, 11, and 12. Full scale boiler tests also were made.

The general plan was to feed the boiler with Naval Academy water (zeolite softened), distilled water or Severn River water contaminated with scale-forming components like calcium sulfate and calcium silicate and to treat with boiler compound according to applicable instructions. Test periods ranged from 70 hours to 200 hours of continuous operation except for weekend shutdowns. Effectiveness of treatment was judged from periodic water tests, corrosion specimens and sample boiler tubes. All results were compared with untreated reference runs. Table 9 is a list of the proprietary treating chemicals used and their approximate compositions.

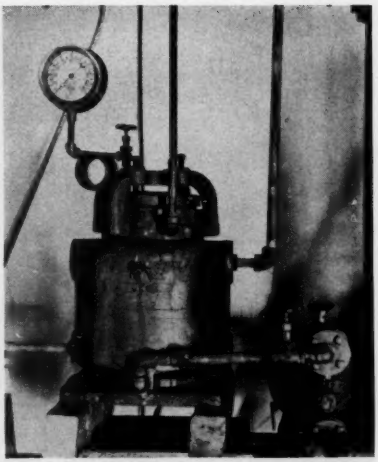


Figure 8. EES Boiler Water Autoclave (1931).

TABLE 9—PROPRIETARY BOILER WATER TREATMENTS
(SOLBERG AND ADAMS, 1933)

Designation of Trade Name	*Composition (Numbers indicate %)**	*Performance*
"A"	4 Fe, 2 Fe_2O_3, 9 FeO, 89 water	Not effective
"B"	42 NaOH, 1 TSP, 1.75 tannin, 1 organic, 54 water	Not effective
"C"	9.5 NaOH, 7.4 TSP, 0.8 tannin, 4.5 organic, 78 water	Not effective
"D"	Graphite, linseed oil and coal tar	Not effective

Figure 11. EES 350 psi Boiler B (1932).

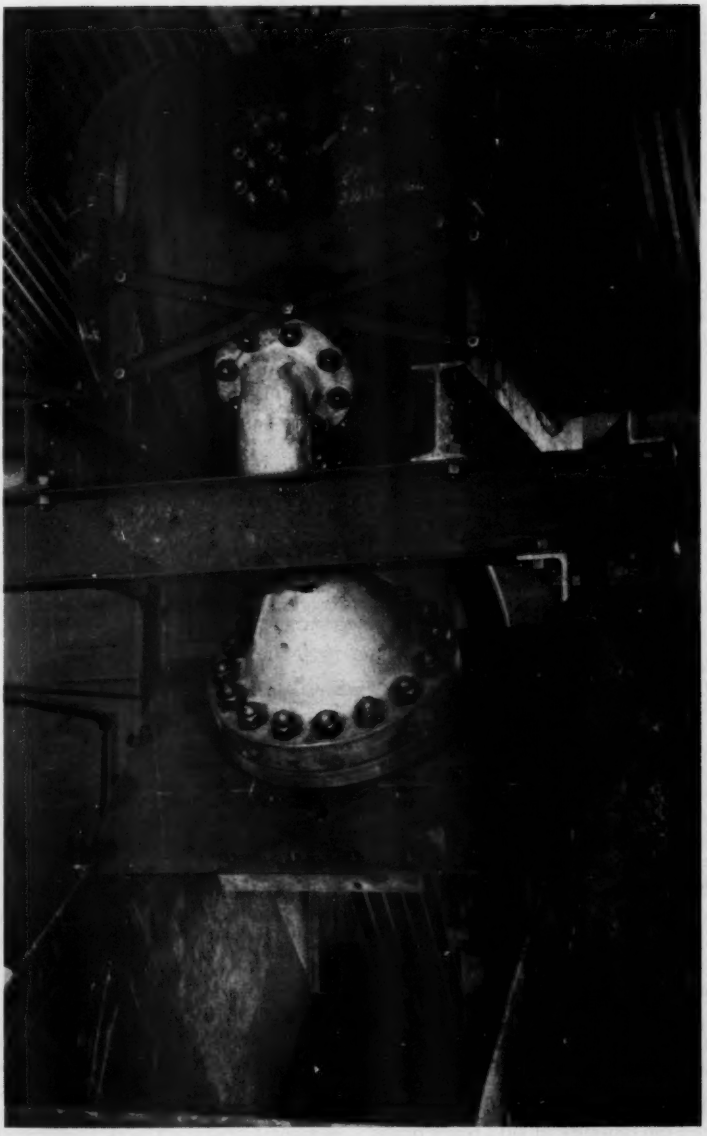


Figure 12. EES 600 psi Boiler C (1932).

"E"	Flaxseed emulsion	Not effective alone but good sludge conditioner
"F"	NaOH and $Na_3P_4O_{12}$ fed separately	Excellent all pressures to 600 psi
"G"	8 NaOH, 12 Na_2CO_3, 5 TSP, 0.3 Tannin 3.5 organic, 72 water	Not effective alone but good sludge conditioner
"H"	Organic and water (caustic phosphate and tannin some times added)	Not effective alone but good sludge conditioner
"I"	None—electrolytic	Not effective
"J"**	A 5.6 NaOH, 52 Na_2CO_3, 1.8 Tannin, 17 Na Al O_2, 25 water plus organic	AB mixture excellent to 350 psi— Neither mixture nor A effective at 600 psi
	B 56.8 Na_2CO_3, 11.3 Tannin, 28 water plus organic	
"K"	33 resin, 47 raw terps, 2 soap, 19 oils	Not effective Caused foaming
"L"	78 disodium phosphate, 10 organic 12 sodium tannate	Not effective (alkalinity too low)
"M"	9 KOH, 4 K_2CO_3 balance water	Examined but not tested

*Organic was sea weed extract in every case.
**A and B chemicals used as one pound balls in bypass on feedwater line.

Analyses of the samples and information from ships showed that water conditions in the Fleet were far from satisfactory. While corrosion was not a serious problem, excessive hardness was traced to the use of the cement tank wash in Table 10.

TABLE 10
CEMENT TANK WASH—1931

Component	*Quantity*
Portland cement	⅔ bucket
Lime	½ cup
Table salt	½ cup

The experimental boiler tests confirmed the inability of 1913 Navy boiler compound to prevent scale. They showed that alkali and phosphate were necessary components of good compounds but that neither was adequate when used alone. Organic additives such as starch, tannin and a host of commercial products were useful only when used in conjunction with these chemicals. Only one of the commercial treatments (Hall) yielded acceptable results at al boiler pressures, though a number showed some degree of promise.

Modified Boiler Compound (1932)

In an effort to correct boiler troubles as rapidly as possible, two major modifications were made before completing the investigation. The use of cement tank wash was prohibited in favor of metallic brown paint Formula 26 and an interim compound (1932) was issued.

TABLE 11
MODIFIED BOILER COMPOUND (1932)

Component	*Percent*
Sodium carbonate	45
Trisodium phosphate (Na_3PO_4 $12H_2O$)	50
Starch	1
Tannin	1
Remainder, moisture and inert materials	

This compound, though markedly better than the existing one, caused several troubles. Manufacturers complained that they could not formulate it from technical grade chemicals because of low tolerances. Forces afloat were unable to obtain supplies of the new compound promptly and had difficulty in distinguishing it from the earlier tannin-bearing material.

In addition to the modified compound shown above several special or stopgap compounds were recommended for specific vessels. The USS *Wyoming* was advised in July 1933 to prepare a facsimile of the anticipated 1933 boiler compound by mixing the following ingredients:

(a) Ten pounds of old Navy boiler compound 1913
(b) Thirty-one pounds of trisodium phosphate (hydrated)
(c) Two pounds of cornstarch

Or,

(a) Ten pounds of modified Navy boiler compound
(b) Fourteen pounds of trisodium phosphate (hydrated)
(c) One pound of cornstarch

In another communication dated 2 October 1933, the USS *Eagle* was advised to use a mixture of 14 pounds TSP and one pound starch for treating boilers fed with Baltimore city water.

Origin of 1933 Compound

This ambitious program led to the conclusions that Navy boiler compound should contain sufficient alkali to maintain the boiler water at 0.4 to 0.7% normal, sufficient precipitating agent to convert all scale forming materials to sludge and maintain zero water hardness and organic additive to fluidize the sludge and minimize foaming. The conclusion regarding alkali and scale preventive was based primarily on the Hall treatment as adapted to sea water systems. Soda ash was used instead of caustic soda because carbonate-phosphate sludges appeared to be less adherent than phosphate sludges alone. Organic was included because proprietary products and laboratory mixtures of this type yielded better coagulation of suspended matter and better fluidity of sludge than did the Hall system.

Why Corn Starch?

The non-proprietary organics tried included several grades of quebracho (tannin extract) and cornstarch. Borax, and inorganic, was tried for the same purpose because of its water softening reputation. Effort was concentrated on starch because of excellent sludge fluidizing and corrosion preventive properties of a proprietary flax seed emulsion (filtralator system) suspected to contain starchy material as its active ingredient. The objectionable color of tannins in water testing also made starch particularly interesting. Cornstarch apparently was not selected by trial and rejection of other starches and dextrins but simply because it was a readily available and well-standardized product. The investigators frequently bought Argo starch from a local grocery store with funds from their own pockets. In response to a Station inquiry the Corn Products Refining Company wrote in June 1933 that while Chemical Abstracts indicated patents and many articles advo-

cating the use of starches, dextrins, potatoes, etc. they could find no information on their reactions in boiler compound nor proof of their usefulness. They expressed the opinion that particles of starch might help in coagulating boiler solids and in keeping deposits softer and more fluid.

Most tests of organics and borax were made in the experimental boilers at pressures from 225 to 600 psi. Additional priming tests were made on cornstarch using the Station's Colven boiler that was particularly liable to carry over. Borax did not improve sludge characteristics at all and showed a tendency toward carryover. Both the tannin and extracts and cornstarch improved sludge fluidity. Tannin yielded maximum effectiveiess at 7% of total boiler compound weight, though it was effective in the range of 3% to 7%. Objectionable color occurred above 2%. Starch was effective in the range of 2% to 7% of the total P_2O_5 content of the compound. The Colven boiler tests showed considerable ability of starch to reduce carryover at high chloride concentrations. In a patent application for starch ingredient of boiler compounds, protection against corrosion and caustic embrittlement also were mentioned. However, no data were presented to support these claims.

It finally was decided that either of the boiler compounds shown in Table 12 would be adequate for Naval service.

TABLE 12
NAVY BOILER COMPOUNDS (1933)

Component *Preferred*	*Percent*
(a) Soda ash	44
Anhydrous disodium phosphate	47
Cornstarch	9
Alternate	
(b) Soda ash	20
Trisodium phosphate (hydrated)	75
Cornstarch	5

Compound (a) was recommended because of the relatively low cost of disodium phosphate and the relatively high moisture content of the trisodium phosphate.

In order to implement the new treatment 10,000 pounds of compound (a) were ordered in the summer of 1933 and instructions were issued to use it as soon as existing supplies were gone. Meanwhile, the Station developed a new soap hardness test (still in use) and modified the nitric acid titration for alkalinity and the silver nitrate titration for chloride. Chapter VI of the Manual of Engineering Instructions was completely rewritten to provide theoretical discussion of water treatment and details of the new boiler compound, equipment and tests. A new standard Navy boiler water testing cabinet was designed (Figure 13) and the first formal purchase specifications were prepared for the cabinet, Boiler Compound 1933 and all of the testing equipment and chemicals. A sample cooler also was proposed, but rejected. Thus within the span of two short years, a scientific boiler compound had been developed and completely implemented.

Fleet trials of Boiler Compound 1933 confirmed its merits, slowly at first, and then by an avalanche of approval. Some vessels, like the USS *Salt Lake City*, accustomed to the thin uniform scales allowed by the old compounds and proprietary treatments mistook decaying scale for new formations and condemned the new formula. Eventually all old scale deposits were removed and everyone was happy. At last the Navy was using a treatment capable of preventing scale as well as corrosion.

Dissolved Oxygen and Deaerators

The clean watersides obtained with the new compound and the steady increase in boiler pressures and heat transfer rates focused attention on dissolved oxygen pitting during the wan-

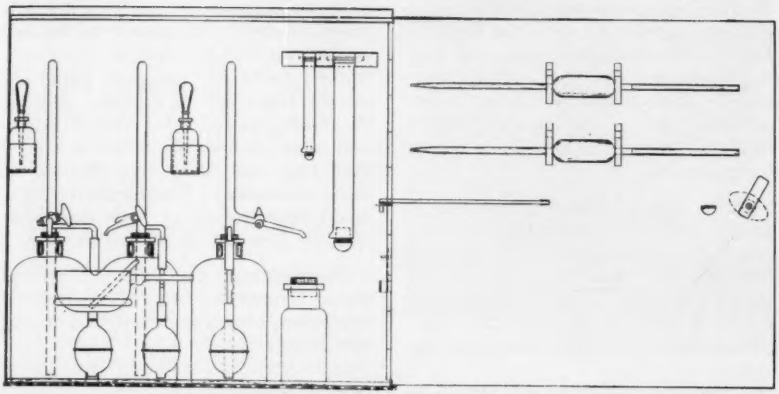


Figure 13. 1933 Boiler Water Test Cabinet—0.1N HNO_3, 0.017N $AgNO_3$, 0.048N Soap and Chromatee, Methyl Orange and Phenolphthalein Indicators.

ing years of the 30's. Dissolved oxygen was not a newcomer. Air corrosion had been mentioned by the earliest investigators. Solberg and Adams made studies of oxygen absorption rates and effects. However, dissolved oxygen remained an elusive and somewhat mysterious contaminant. The Station launched a program to develop better test methods for dissolved oxygen and to evaluate the various types of feedwater systems from the standpoint of deaeration efficiency. The first serious shipboard survey was made by the personnel of the USS *Arkansas* under the direction of R. C. Adams during a midshipmens' cruise in the summer of 1935. Regular samples from five water sources in the open feedwater system showed dissolved oxygen concentrations up to 5 ppm and a positive correlation between the percentage of makeup feedwater and concentration of dissolved oxygen in the feedwater. A typical data table is shown below.

TABLE 13—DISSOLVED OXYGEN SURVEY DATA USS ARKANSAS—JULY 1935

Day	Steam 1000 lb./hr	% Make-up	Temperature °F Hot Well	Temperature °F Main Feed	Dissolved oxygen content, ppm Main Feed	Hot Well	Main Condensate	Low P Drains	Make-up	Evap Drains
1	136	4.5	—		3.9	—	.04			
2	231	5.5	—	184	3.2	—		2.1		
3	229	5.5	—	184	3.1	—			5.0	
4	231	6.8	—	184	3.0	—			5.0	
5	176	6.3	126	192	2.9	1.4				1.3
11	319	2.6	—	210	4.0	—	0.8			
12	114	15.6	—		3.9				5.0	
13	37	16.2	—		3.6	2.5				
14	48	21.5	122		2.6			1.4		
29	95	10.7	—		3.2	2.7				
30				218	3.2			2.5		2.0

This survey proved that dissolved oxygen concentrations were too high for practicable chemical oxygen scavenging. The Station recommended reduction to approximately 0.1 by mechanical improvements like drowning the discharge line from the loofa sponge compartment.

Other shipboard test of open, semi-closed (surge tank floating on line) and vacuum closed feedwater systems followed the *Arkansas* survey. The vacuum system proved to be impracticable because of air leaks. The semi-closed system, though reliable and apparently adequate for medium and low pressure work, left much to be desired in the high pressure range. Feedwater dissolved oxygen generally ranged from 0.05 to 0.5 ppm. During the closing years of the decade attention was turned to pressure closed flash type and spray type deaerating feedwater heaters. These tests left little doubt that the spray type deaerator was the best for shipboard service.

The electrical salinity indicator which formed the basis of our present equipment also was introduced during this decade.

THE 40'S—PERIOD OF TRANSITION

This period, still clearly visible over the horizon of Naval history, was characterized by modernization of terminology and specifications; improvement of test methods, boiler record systems and personnel training programs and provision of special treatment for unusual feedwaters and prototype high pressure boilers. There was a brief time out in 1941 to revise the boiler compound specifications as shown in Table 14 on the basis of service data from the fleet and to change the alkalinity limits of 0.25 to 0.35% normal. This was modified in 1947 to insure better mixture.

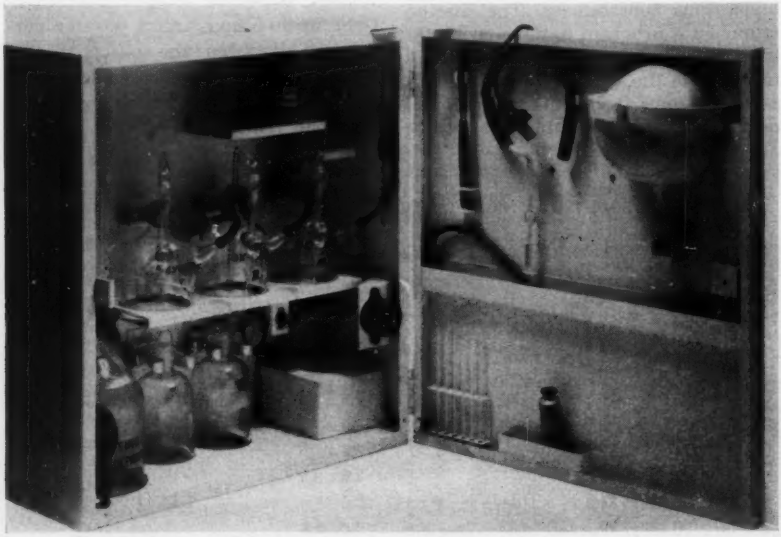


Figure 14. First Standard Dissolved Oxygen Test Cabinet (1942).

TABLE 14
NAVY BOILER COMPOUNDS OF THE 40's

Component	*Limit 1941, Min. %*	*or Range 1947*
Disodium phosphate	.47	45 min
Sodium carbonate	...35.7	—
Starch	.....11.4	—
Ratio, Na_2HPO_4/Na_2CO_3	.—	1.29 to 1.37
Ratio, Na_2HPO_4/Starch	..—	3.89 to 4.11

The early years of the decade were devoted primarily to intensive search for improved dissolved oxygen tests. This resulted in a referee procedure capable of detecting dissolved oxygen down to 1 part per billion or less as well as a simplified shipboard method suitable for the range of 20 parts per billion or more. The referee procedure was accepted as the standard of the American Society for Testing Materials. Numerous investigators have confirmed its superior accuracy and precision since that time. A standard shipboard dissolved oxygen test cabinet also resulted from these studies (Figure 14). Figure 15 is the laboratory deaerator used in development work.

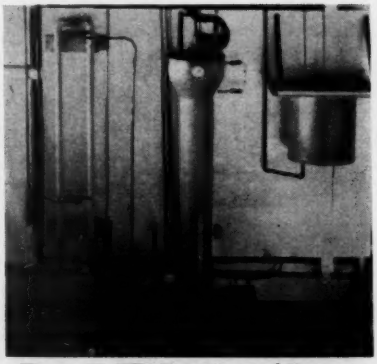


Figure 15. EES Experimental Deaerator (1941).

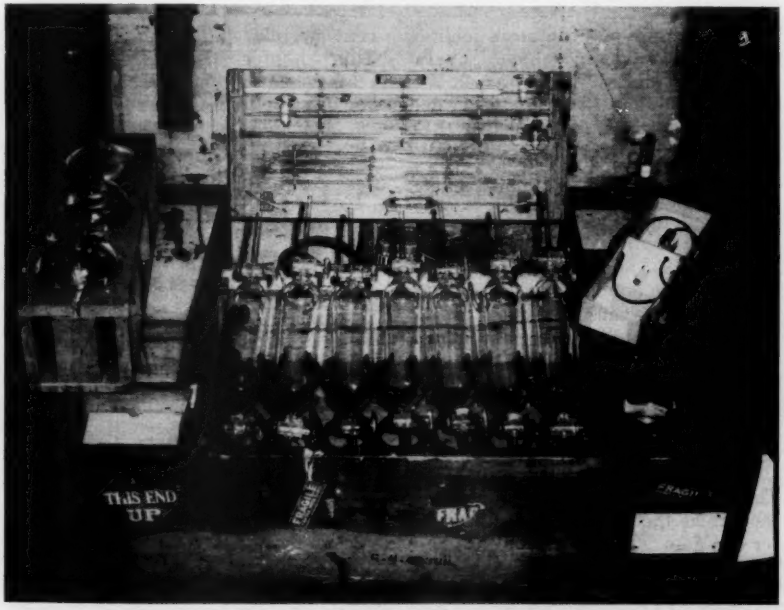


Figure 16. EES Deaerator Test Kit.

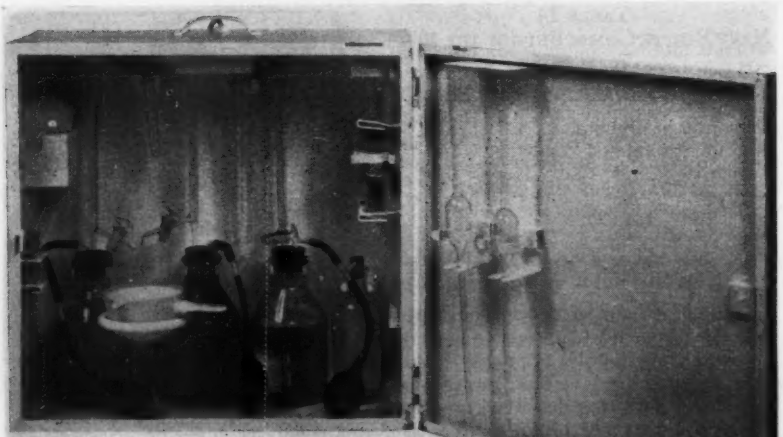


Figure 17. Latest Boiler Water Test Cabinet—0.05N, HNO_3, 0.025N $HgNO_3$, 0.01N Soap and Diphenylcarbazone, Methyl Purple and Phenolphthalein Indicators.

Many prototype deaerators for new fighting ships were evaluated by the referee oxygen test during World War II. Figure 16 shows a travel-weary dissolved oxygen test kit carried by EES personnel to various ships and manufacturers' plants for this purpose. Many of the vessels which participated in the Program long since have gone to Davy Jones Locker. The program reached its climax in 1943 and 1944 with papers by Adams, Barnett and Keller on dissolved oxygen testing and by Adams on approval testing of deaerating equipment.

The revised boiler compound and new dissolved oxygen tests required another revision of the Chapter on boiler feedwater treatment (now Chapter 56, Bureau of Ships Manual). The Station siezed this opportunity to modernize the units for reporting water test data. The revision published in 1948 used the scientific terms equivalents per million and parts per million instead of the obsolete terms per-cent normal and grains per gallon. This chapter also introduced new reagent strengths (0.05N HNO_3 and $AgNO_3$ and 0.01N soap), the new alka-

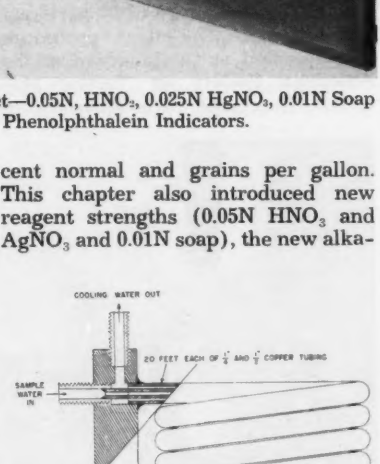


Figure 18. Shipboard Water Sample Cooler (Proposed 1948).

linity indicator, methyl purple and a new boiler water test cabinet (Figure 17). For the first time, it included instructions for a dissolved oxygen test and the drawing of a water sample cooler (Figure 18).

Records and Programs

The relatively large number of ships and waterside boiler problems during the War years account for the work on improvement of records. Standard EES test reports first were supplemented with looseleaf summary ledgers and cross-catalog card files. Eventually these were replaced with IBM punch card records to facilitate rapid analysis of water side statistics. This system, still in effect at the Station, makes it possible to learn in a few minutes what vessels, what parts of the power system, and what specific water problems are most pertinent.

The improved records confirmed the adequacy of standard Navy water treatment and pressure closed deaerating feedwater equipment. They showed that misunderstanding of treatments and tests, faulty equipment and sluggish supply lines were by far the most important sources of waterside problems. In view of these findings, the Station initiated a campaign to improve indoctrination, training and trouble shooting. A field survey system proposed in 1945 was not approved as a formal, financed project though it was established on an informal basis as part of the overall development program. It has functioned well through cooperation with the training schools, particularly Deslant Afloat Enpineering School, Newport, Rhode Island, the training code of the Bureau of Ships (258), the East Coast Yards and the publication facilities of the Bureau of Ships and the Society of Naval Engineers.

Treatment and Test for Prototype Boilers

After World War II came new types of Naval steam power plants and with them came new water problems. High pressure boilers like those of DD828 and DL's 1 to 5 (up to 2000 psi) demanded closer control on feedwater quality and precise, heat stable treatment not obtainable with Navy boiler compound. In 1947, the Bureau of Ships asked for a new chloride test capable of detecting 0.06 epm of chloride (approximately 2 ppm). In 1948, a special treatment for Naval high-pressure boilers was requested, and finally in 1950, the Station was asked to recommend water treatment for the first submarine boiler (STR).

The improved chloride test, which at first seemed hopeless, was developed during 1947 and 1948, reported to the Bureau in 1949 and established as the standard shipboard test in 1950. It consists of a mercuric nitrate titration using mixed indicator of diphenylcarbazone and bromophenol blue. It is capable of detecting 1 part per million or less. It has the advantage over the superseded silver nitrate-potassium chromate test of essentially zero blank and sharper end point color change (yellow to blue violet). This was the first real change in the chloride test in 40 years.

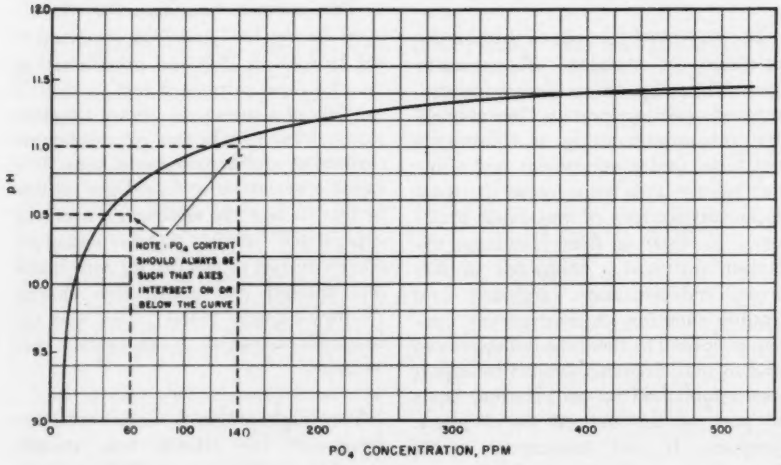
The high-pressure boiler water treatment for DD828 was recommended in December 1948 and approved by the Bureau in 1949. It was service tested from 1949 to 1951 with excellent results, and still is in effect. The treatment consists of precision control with dipotassium phosphate and potassium hydroxide to maintain the boiler water conditions in Table 15.

TABLE 15—BOILER WATER LIMITS, DD828 AND DL's 1 TO 5

Component or Property	*Limit or Range* DD828	DL's
pH	10.6 to 10.8	10.4 to 11.0
Phosphate, PO_4, ppm	5 to 20 (prefer 5 to 10)	10 to 25
Chloride, Cl,epm	——	10
Total Solid, ppm	1000 max.	——

No organics were included because of their instability at the 2,000 pound maximum operating pressure. The narrow control limits are attained by adding the treating chemicals separately instead of as a mixture. They insure minimum solid in the water and precipitation of magnesium as silicate or hydroxide rather than stubborn magnesium phosphate sludge. Potassium salts, though more expensive than sodium salts, are more fluid and soluble and therefore less liable to deposit (hide out) in hot spots. Hide out of treating chemicals is a pertinent problem at high pressure and heat transfer rates.

The principal disadvantage of precision control from the standpoint of Naval service is the relatively low reserves of alkali and phosphate (hardness preventive). Sea water leakage in the feedwater system, if not detected promptly, could wipe out these reserves and cause scale or corrosion.



pH	10.5	10.6	10.7	10.8	10.9	11.0
PO_4, Min.	40	50	60	75	95	120

Figure 20. Boiler Water Control Chart (STR Boiler).

Figure 19. Commercial Phosphate Color Comparator (DD828).

Another possible source of trouble is that boiler water carryover might leave molten potassium compounds in the superheater and cause caustic corrosion. Tests for pH and phosphate are made with color slide comparators like that in Figure 19. Only the chloride test uses standard Navy test equipment. The color tests consist simply of adding certain reagents to the water sample and comparing the resulting color standards. Reagent instability and color interferences problems still have not been eliminated completely.

The boiler water treatment on the DL vessels is essentially identical with that on DD828 except that sodium salts are used instead of potassium salts. The STR submarine boilers are treated with the same chemicals, disodium phosphate and caustic soda, using the coordinated pH-phosphate control plan of Whirl and Purcell. Here the chemicals are added separately to insure intersection of the pH and PO_4 concentrations below the curve in Figure 20. Boiler water of this quality prevents caustic attack by precipitating trisodium phosphate instead of caustic in hot spots. It has the disadvantages of promoting hideout and heavier sludges, particularly magnesium phosphate. These are not pertinent at the operating conditions of the STR boilers. It is more flexible than boiler compound treatment and provides more chemical reserve than precision control. The colorimetric tests are the same as those described above. This treatment has provided good protection to submerged parts of the prototype stainless steel boilers.

The first tests of a dissolved oxygen meter (Cambridge) were made in conjunction with prototype boiler tests. The meter, shown in Figure 21, scrubs dissolved oxygen from the water sample with pure hydrogen and then measures it by comparing the thermal conductivity of the gas mixture with that of pure hydrogen. Figure 22 compares a typical electronic strip chart record of this instrument with referee dissolved oxygen data. The meter responds quickly and measures dissolved oxygen accurately. The principal shortcoming is that it recovers slowly from slugs of dissolved oxygen. This can be overcome by an automatic zeroing device now available. Shock resistance should be similar to that of other electronic equipment. A shipboard trial has been recommended to the Bureau of Ships.

THE LIVING PRESENT

The majority of Naval boilers still are treated with standard Navy Boiler Compound 1951, the second modification of the 1941 mixture.

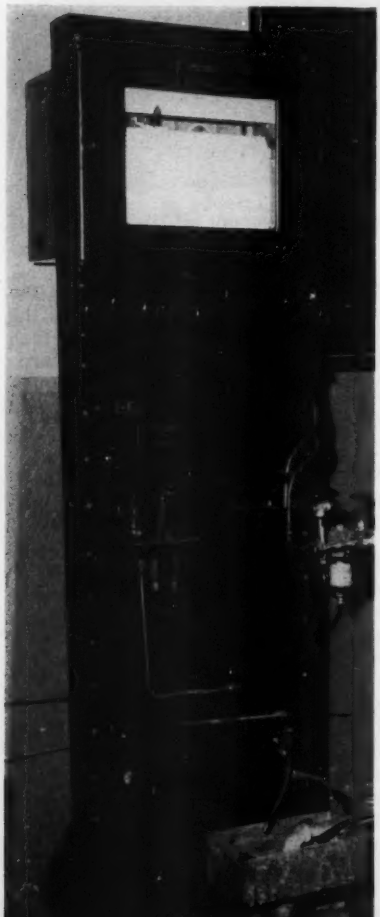


Figure 21. Cambridge Dissolved Oxygen Meter.

The last three years of Navy water technology have been devoted primarily to special water treatments and to unusual water tests associated with them. Adoption of nonstandard treatments for DD828, the DLs and the STR boiler during the past decade, represented a change in the philosophy that Navy boilers should be treated on the average, disregarding the extremes. The LSD and CVE vessels equipped with reciprocating engines and diatomaceous earth feedwater filters represent such an extreme. Their feedwater filters contaminate the boiler water with relatively large amounts of calcium sulfate silica and aluminum compounds. This requires large quantities of boiler compound and results in excessive sludge and alkalinity. In March 1952, Station engineers started a corrective treatment on the USS *Corregidor* (CVE58). This consisted of feeding caustic soda (lye) and disodium phosphate-cornstarch mixtures separately as necessary to maintain the boiler water conditions shown in Table 17.

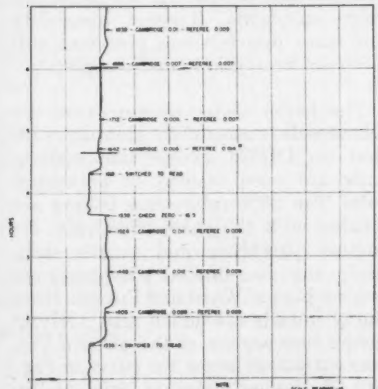


Figure 22. Cambridge Oxygen Meter Record.

TABLE 16—BOILER COMPOUND (1951)

*Component or Property	Limit or Range
Moisture	6% Maximum
Ratio Na_2HPO_4/Na_2CO_3	1.16 to 1.47
Ratio Na_2HPO_4/Starch	3.70 to 4.20
Total percentage of ingredients	97.5 to 101.5

*This is essentially 4 parts Na_2HPO_4, 3 parts Na_2CO_3 and one part starch.

TABLE 17—BOILER WATER LIMITS DIATOMITE FILTER VESSELS (LSD 1 TO 8 AND SOME CVES)

Component or Property	*Limit or Range*
Alkalinity, epm	2.5 to 3.5
Phosphate, PO_4, ppm	20 to 100
Chloride, Cl, epm	15 max.

The alkalinity limits and test are the same as those on vessels using boiler compound. Phosphate is determined instead of hardness using colorimetric equipment like that on DD828. Five percent of the water is blown to waste for every pound of treating chemical added in order to remove the large quantity of sludge which results from feedwater contamination.

Before introduction of this split treatment, the watersides of the USS *Corregidor* accumulated heavy deposits resembling the meat of a cocoanut in color and thickness. Heavy sludging and high alkalinity troubles have disappeared since its initiation.

The same treatment was started on LSD Vessels 1 to 8 immediately after the trials on the USS *Corregidor*. The same excellent results have been obtained and the stubborn calcium sulfate scales and blistered tubes which used to plague these vessels have disappeared.

A special chemical treatment for preventing dissolved oxygen corrosion also is being tried on the USS *Robinson* (EDE-220) and the USS *Coolbaugh* (DE-217). This consists of using sodium sulfite (Na_2SO_3) in addition to the regular Navy boiler compound so that a minimum of 25 ppm sulfite (SO_3) is in the boiler water at all times. Sodium sulfite reacts with dissolved oxygen to form disodium sulfate (Na_2SO_4) and water. The sulfite test uses standard navy glassware, but requires new chemicals. The treatment is intended to minimize oxygen pitting during intermittent operation of boilers with open or semi-closed feedwater systems. No difficulty has been encountered with the treatment and tests, and dissolved oxygen pitting, has been reduced particularly in desuperheater piping. There is some indication that the relatively high concentration of sulfite has caused it to carry-over and corrode the superheaters. Unfortunately, test specimens removed to check this point were lost in transit. A further disadvantage is that seven pounds of sodium sulfite, a white salt, are required to react with one pound of dissolved oxygen. This can lead to excessive boiler water solids and excessive blow down requirements.

Laboratory and full scale boiler tests of the glamorous new oxygen scavenging chemical hydrazine also have been underway for two years. This ammonia like colorless liquid has the advantage over sulfite of combining with oxygen pound for pound, and

of yielding water and nitrogen as its only reaction products under optimum conditions. Therefore it contaminates neither the water nor the steam if handled properly. Unfortunately, if controlled improperly it can decompose to ammonia and cause condensate corrosion in non-ferrous systems. Hydrazine also forms a very interesting salt, hydrazine phosphate. With this it may be possible to provide both phosphate (hardness protection) and oxygen scavenging in a split alkali phosphate system such as those already described. Bench tests with this chemical have shown it to be promising for prevention of corrosion in cold aerated water like that in a secured boiler. Time will tell if these interesting chemicals are adaptable to use by the Fleet.

The latest and most puzzling treatment problem is that of preventing stress corrosion in stainless steel boilers of the STR and mine sweeper types. This is aggravated by many puzzles. For example, stressed parts, such as rolled tube ends and U-bends will not corrode when immersed in treated boiler water but do corrode when exposed in the steam space above such water and wetted with it periodically. Chloride, a water phase constituent, appears to be a contributing factor; however, there is evidence that oxygen also takes part in the reaction. To date, neither vapor phase inhibitors, like the waxy amines nor water phase inhibitors like chromate have reduced the cracking. Figure 23 shows one of several autoclaves that EES is using to investigate this problem.

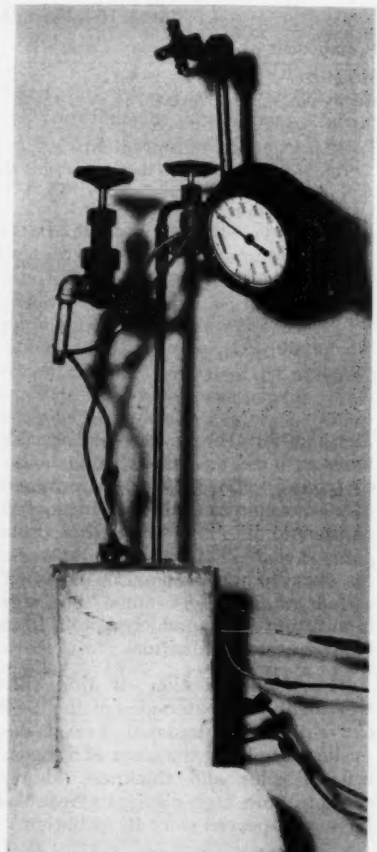


Figure 23. Autoclave for Stainless Steel Cracking Tests.

Latest Word in Testing

There never has been a time when fleet water tests were completely adequate. Complexity of equipment and techniques, instability of reagents from the standpoint of shipboard supply and obscurity of end point signals are common shortcomings. These deficiencies are most pronounced in the present tests for hardness and dissolved oxygen. The soap hardness test requires a five-minute waiting period which is very trying on the patience of an operator who consistently has recorded zero many times before. The test is liable to false end points in the presence of magnesium salts and reads traces of heavy metals as hardness. Search for a better hardness test was started in 1949. During the past three years very promising progress has been made in adapting the newly developed Schwarzenbach complexometric titration to Navy use. Contrary to its title, this is a very simple test. A liquid indicator turns the sample red if hardness is present and titration to a blue color with a second chemical measures the hardness. The test is sensitive enough to detect 1 ppm of calcium or magnesium hardness. The color change is sharp and distinct. Recognized interferences such as manganese, copper, and zinc generally are not pertinent in Navy boiler water. The only real shortcomings is instability of reagents. Effort has been concentrated on this problem and to date stable forms have been obtained for two of the three chemicals required. Informal trials of the method have been made on several ships and formal trials recently were recommended to the Bureau.

The existing shipboard dissolved oxygen test has many disadvantages. Two of the four reagents are extremely corrosive and one causes continuous trouble with frozen stopcocks. It will not detect oxygen below 0.02 ppm, though a modern deaerator is guaranteed to supply water of 0.014 ppm or less. A long search for better oxygen methods finally was rewarded in 1952 with discovery of a promising indigo carmine colorimetric procedure. Basically this consists of adding a mixed reagent to the water sample, shaking it and comparing the color with a series of standards. The water sampling and mixing procedure is simple but unique and employs the sampling bottle sketched in Figure 24. The mixed re-

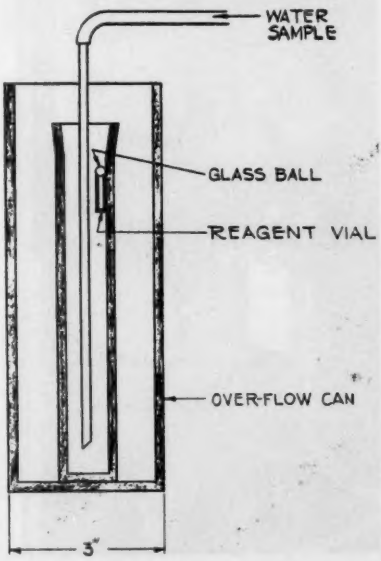


Figure 24. Sampling Tube, Colorimetric Dissolved Oxygen Test.

agent of indigo carmine, glucose, caustic soda, glycerine, and water is pipetted into the small vial inside the sample tube. The vial is sealed with a glass bead and the sample water is flowed into the bottom of the tube and overflowed past the vial of reagent until the tube is thoroughly flushed. The stopper then is wetted and inserted, and the tube is inverted several times to mix the reagents and develop the color.

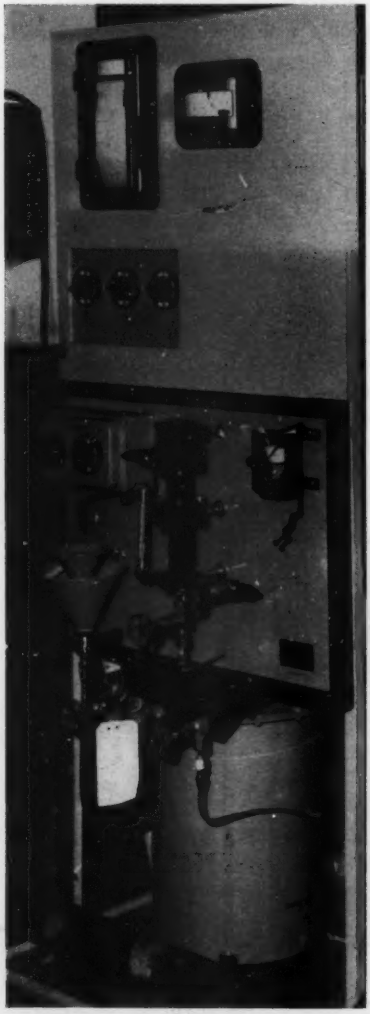


Figure 25. Hartmann-Braun Dissolved Oxygen Meter (German).

Numerous laboratory and service trials of this method have shown it to be as sensitive and almost as reproducible as the Station's referee method for dissolved oxygen. Instability of the mixed reagent again is the major problem. Attempts are being made to provide the necessary chemicals as two or three stable components with simple directions for mixing prior to use. Perfection of this method would make it as easy to test for dissolved oxygen as to test for alkalinity. A Hartman-Braun oxygen meter (German) also is being tested (Figure 25).

Needs for other special tests have grown out of the special treatments discussed earlier. For example, the cracking of stainless steel boiler parts resulted in the need for a microchloride test. (0.5 ppm and lower). The Station has adapted the standard mercuric nitrate titration to this need as an interim measure and now is working on more sensitive mercurimetric spot tests and colorimetric tests.

Adoption of hydrazine as a water testing chemical also will involve some puzzling test problems. The small concentrations involved (approximately 0.05 ppm) are difficult to determine by any means.

In 1953 a simple shell and coil heat exchanger finally was adopted as standard equipment for cooling boiler water aboard ship. The cooler can also be used for collecting dissolved oxygen test samples if the sea water temperature is below 70°F. It is shown in Figure 26.

WHERE ARE WE HEADING?

Prophecy in the field of science always has been a risky business. The trend in the Fleet seems to be toward high pressure boilers of 1000 psi or more. Such boilers will be treated by split alkali phosphate systems like those already described. As the number of these boilers increases, it will be less profitable and logical to retain Navy boiler compound. The increasing number of metal oxide sludging problems also points to the need for a treatment less liable to contaminate the steam with carbon dioxide. Substitution of split treatment for boiler compound treatment of course would mean introduction of new test equipment. In all probability, the boiler water cabinet in its present form will disappear in favor of a smaller cabinet with color comparators for several tests (chloride excepted). There is little doubt that an oxygen scavenging treatment of some type will be used at last on the vessels that are particularly vulnerable to dissolved oxygen damage. It would be foolish for anyone to guess what treatments might eventually be required by critical pressure boilers (3500 psi or higher) and future types of thermal reactor boilers. It is a better policy to meet these problems as they arise and to hope for the best.

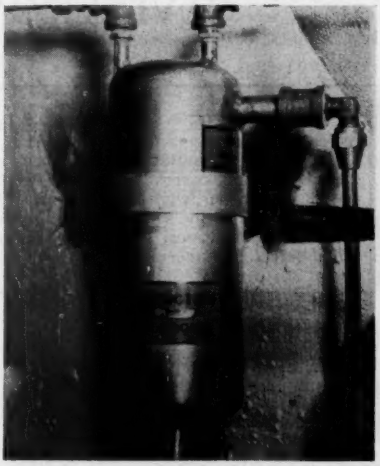


Figure 26. First Standard Shipboard Water Sample Cooler (1953).

CONCLUSIONS

One need not look back on this half century with regret or feeling that progress could have been better. Frank Lyon's boiler compound certainly resembles modern boiler water treatment as closely as the 1910 boiler resembles its counterpart in the present generation. It would be comforting to think that the first 50 years were the hardest, but I doubt it. Whether or not Navy water technology will retain the same high level of proficiency and demand the same respect from posterity 50 years from now will depend to a large extent on perpetuation of attitudes and programs like those described in this paper. Let us hope that all concerned will meet this challenge.

PRESENT STATUS OF CAVITATION RESEARCH

ACKNOWLEDGEMENT

The effect of cavitation on performance of hydraulic devices, and losses from mechanical and corrosion damage, have been subject to intensive study for many years. This paper, written by Robert T. Knapp of the California Institute of Technology, reviews the scope and findings of such studies and suggests the course of future investigations. The paper was published in the September 1954 issue of *Mechanical Engineering*.

In 1947 the David Taylor Model Basin published an annotated bibliography of cavitation (1). The individual annotations make it possible to construct a general development outline of the cavitation field. A few of the interesting high lights are discussed in this paper.

It is a common impression that the hydrodynamic disease of cavitation has been recognized only quite recently. Hence it is a shock to observe that in 1754, in discussing hydraulic turbines, Euler recognized the possibility of cavitation, although he did not call it by that name. However, the nineteenth century was not very productive of articles, since the first one appeared in 1865, and only 18 are found in the 35 remaining years in that century.

The next 20 years showed a definite increase in cavitation interest, since 42 articles appeared, about four times the rate of the previous period. However, during these 20 years the average annual production did not vary appreciably. The big change in interest came about 1920, since the following 5-year period averaged nine articles per year, and the next three 5-year periods averaged 24, 36, and 48, respectively. The latter period, ending with 1939, is an all-time high. From 1940 to 1944 the yearly average dropped to 18. The last three years covered by the bibliography averaged only 12 articles. Two factors which explain this great drop are (*a*) the interruption caused by the outbreak of war, and (*b*) the fact that some aspects of cavitation information were recognized as having military value and were withheld from publication.

SOURCE AND SCOPE OF PUBLICATIONS

Eighty per cent of all articles originated in the United States, United Kingdom, Germany, and Switzerland. Of this group 20 per cent are from the United Kingdom, and 40 per cent from the United States. The United King-

dom showed the greatest early activity. Indeed, the first article published in the United States was in 1899. Up to 1920 the United Kingdom published more articles on cavitation than the combined output of the other countries.

Two other classifications are interesting: [1] Subject, i.e., cavitation process, effects of cavitation on performance, and damage; [2] approach, i.e., theory, description, and experiment. For these classifications the listings exceed the number of articles, since many articles contain more than one subject or approach. The interest in subject matters, as indicated by the number of articles, is (*b*) cavitation process 211, (*b*) performance effects 490, (*c*) damage 273. From the viewpoint of approach the totals are (*a*) theory 252, (*b*) description 235, experiments 487. Two facts stand out; the interest in performance effects is equal to the combined interest in the cavitation process and in damage; interest in the experimental approach is equal to the combined interest in theory and description. Interest in both the cavitation process and effects on performance developed as soon as the phenomenon was recognized; whereas damage is hardly mentioned before the twentieth century.

MECHANICS OF CAVITATION

An attempt will be made to evaluate the present state of knowledge in different areas of the cavitation field and to outline some of the more important problems for future investigation. It should be understood that this does not purport to be a consensus of the workers now active in the field but is simply an expression of the current opinion of the author.

It is now generally recognized that cavitation is a dynamic process involving the formation and collapse of vapor-filled cavities in a flowing liquid. With normal liquids, these cavities form where the local pressure would drop below vapor pressure if the cavity did not form. Conversely, collapse commences when a vapor cavity is transported to a region where the pressure is above vapor pressure. There are several characteristic types of cavitation. In one common type the cavity remains fixed with respect to the boundary surface. The liquid-vapor interface is very active, with material interchange between the two phases, presumably with vaporization in the upstream regions and condensation toward the downstream end. There is physical entrainment varying from a smooth, fine-grained process to a violent large-scale process in which a major fraction of the cavity may be suddenly swept away, followed by regrowth and resumption of entrainment. There is a relatively rapid circulation within the cavity, induced by the flowing liquid.

In another distinctive type of cavitation, discrete traveling cavities are formed. They move with the local liquid velocity. Such cavities have been studied in detail (2) for cases in which the pressure distribution in the noncavitating flow is known. The cavity first appears where the pressure drops to vapor pressure. It grows as long as the pressure is at or below vapor pressure, and starts to collapse as the pressure becomes greater than vapor pressure. Experimental measurements of the time rates of growth and collapse of such isolated spherical cavities agree well with the behavior predicted by Rayleigh (3). Deviations are in the direction to be expected from the idealized boundary condi-

tion used by Ralleigh, e.g., nonviscous incompressible liquid, completely empty cavity. In this type of cavitation, as the intensity is increased, the number of cavities per unit area increases rapidly until there is interference between them. During the growth period two or more may coalesce, and in the collapse zone they often collapse in tandem, i.e., the first in a series collapses into the second, and so on. Although such interference makes analysis impossible, it is obvious that the general mechanics is the same as that for the solitary traveling spherical void.

Another distinctive type of cavitation has been observed so frequently at the tips of propeller blades that it is called "tip" cavitation. The cavity forms in the core of a vortex and appears as a long small-diameter tube. On revolving propellers these tubes appear as helices. They trace the path of the propeller tips relative to the liquid and hence do not conform with the flow lines of the liquid.

Occasionally other apparent types of cavitation have been observed. Usually a little study shows that they are combinations of the three types just described.

EFFECTS OF LIQUID PROPERTIES ON CAVITATION

The pressure relations just described are typical of cavitation in normal liquids, and in relatively large-scale installations such as pumps, turbines, or field structures, in general. However, under some conditions the properties of the liquid may affect the cavitation parameter* for incipient cavitation and possibly influence the type of cavitation. These properties fall into two groups: (*a*) Those of the pure liquid, and (*b*) the effect of impurities. In bubble or void formation in a pure liquid, the first property that must be considered is tensile strength. It is commonly assumed that liquids have no tensile strength and yet both theory and experiment agree that pure homogeneous liquids have high tensile strength, tens of atmospheres or higher. If the tensile strength is effective, it will decrease the K for incipient cavitation. If the liquid shows little or no tensile strength, there must be flaws in the liquid, due either to the presence of impurities or to holes in the liquid structure itself. The possibilities of the latter are determined by the molecular structure of the liquid. Current theories indicate that the probable number of holes per unit volume is far too small to explain the observed cavitation characteristics. This tends to focus attention on the impurities.

The effects of impurities in producing weak spots have not been explored completely. However, some general remarks can be made. If the impurities are completely dissolved gases, there is good experimental evidence (4) to show that they have little practical effect. For example, water saturated with air at atmosphere pressure exhibited a tensile strength of approximately 500 psi when all of the air was completely dissolved. Conversely, gases present in the gaseous phase, e.g., bubbles, are obvious flaws in the liquid since they supply gas-liquid interfaces and thus reduce the

*The cavitation parameter is

$$K = (p_L - p_c) / \rho\left(\frac{V^2}{2}\right)$$

where p_L is the pressure in the undisturbed liquid, p_c the pressure within the cavity (usually the vapor pressure), V the relative velocity, and ρ the liquid density.

apparent tensile strength of the liquid to zero. If the impurity is a completely miscible liquid, it would appear to be comparable to a completely dissolved gas. If the impurity is a dissolved solid, it could be expected to fall in the same category as a dissolved gas or a miscible liquid and presumably would have little effect on the cavitation properties of the solvent. If the impurity is an insoluble solid, then each solid-liquid interface is a flaw whose relative weakness depends upon such properties as wettability, bond strength, size, shape, and so on.

Once a tiny cavity is formed, it is clear that the thermo-dynamic properties of the liquid (latent heat, specific heat, and heat conductivity) will affect the rate of growth of this cavity. It should be noted that the growing interest in the properties of the liquid and its impurities is not merely academic, since these factors seem the most likely ones to explain both the differences in cavitation behavior shown by different liquids and the cavitation "scale effect" whose existence is becoming recognized.

EFFECTS OF CAVITATION ON PERFORMANCE

Like many other aspects of cavitation, its effects on the performance of hydraulic devices have long been recognized but only partially understood.

These effects manifest themselves in at least three different ways:

(*a*) *An Increase in Resistance to Flow.* In pumps this effect may appear as an increase in power requirements. In turbines it may manifest itself as a drop in power output for a given head and flow rate. One possible explanation of this increase in resistance is that it is the net result of two opposing factors, a decrease in skin friction on the surface covered by voids and an increase of form resistance as the effective shape of the surface is changed by the presence of the voids. Thus in some hydraulic machines a slight increase in efficiency has been observed at the inception of cavitation, followed by a drop as cavitation develops. This implies that in the early stages, the decrease of friction more than balanced the increase in form drag.

(b) A Major Change in or a Breakdown of Flow Pattern. This effect manifests itself as an upper limit of the flow rate irrespective of an increase in the pressure drop which causes the flow.

(c) *Development of Vibration.* The "spoiling" of the flow by the presence of cavitation frequently manifests itself in premature separation, the formation of Kármán vortex-streets, and similar pulsating-flow phenomena.

MECHANICS OF CAVITATION DAMAGE

In this phase of the cavitation phenomenon there is more speculation than knowledge. One basic reason for this is that since cavitation damage is caused by the cavitation process, it is extremely difficult to understand the mechanics of cavitation damage without first understanding the mechanics of the cavitation process. Thus it is little wonder that this field is still largely unexplored. However, some landmarks have been charted by the individuals who have ventured into this area.

(*a*) Cavitation can damage solids

by purely mechanical processes under conditions in which chemical action is impossible. For example, glass, quartz, stainless steel, stellite, gold are damaged by cavitation in pure water. Metallographic studies show clearly that metal surfaces damaged by cavitation show evidence of mechanical working.

(b) In many cases cavitation seems to stimulate corrosion. Experiments indicate a critical intensity of cavitation for each material below which little or no detectable damage occurs. Above this intensity threshold, the rate of damage increases with the intensity. The situation for corrosion is similar; i.e., a given material may resist the action of given acids or salts if their concentration remains below the threshold values; whereas, if concentrations are increased above these values, appreciable corrosion damage may result. However, the combined effect of cavitation plus corrosion may produce damage under conditions in which either one acting alone does not. It is the author's opinion that this possibility of combining effects is responsible for some of the confusion now existing in the literature on cavitation damage.

RELATIVE RESISTANCE OF CONSTRUCTION MATERIALS TO DAMAGE BY CAVITATION

In this phase of cavitation there is much empirical information. Some of it is direct observations of various materials under cavitating conditions in hydraulic structures and machines. The material may be either an integral part of the equipment, or applied as a protective overlay. Many publications discuss the merits of different methods of application of the protective layer and evaluate the relative resistance of different materials. The consensus is that the best resistance to cavitation damage is exhibited by hard materials with high elastic and corrosion limits, e.g., stainless steel and stellite.

Rubber is an interesting exception. In some equipment a layer of rubber, properly cemented or vulcanized, shows practically no damage under conditions which damaged the metal surface. However, if the intensity of cavitation is increased above some critical value, the rubber protective layer is destroyed rapidly, usually coming off in large units, and an examination shows evidences of high internal temperatures. This is consistent with the concept that cavitation damage occurs by mechanical working. Since rubber has relatively high hysteresis and low heat conductivity, its internal temperature should rise under the effect of cavitation, and if the energy dissipated through hysteresis becomes great enough, its temperature rise could produce physical damage.

Many attempts have been made to classify the resistance of materials by means of laboratory tests. In this country the most used is the magnetostriction method. In Europe the impingement of a stream of individual drops of test liquid also has been widely used. The relative resistances found by different methods do not always agree nor do the results of field use necessarily rank the material in the same order of merit. There are many possible reasons for these discrepancies. There is no conclusive evidence that laboratory tests actually simulate damaging conditions in field structures and machines. Also, it is not certain what are the significant parameters in these tests, e.g., surface finish and intensity of the cavitation.

In the author's opinion the latter is a fruitful source of difficulty.

The normal magnetostriction technique is to establish a given frequency and amplitude of vibration of the specimen and to observe the loss of weight as a function of time. Hence, if this test does produce conditions that are typical of cavitation, they are certainly only of cavitation at one given intensity. As previously stated, there is evidence that different materials have different damage thresholds; however, little is known about the behavior of materials having the same damage threshold. Thus it would be desirable to make tests of the materials at the cavitation intensities encountered in the application under consideration. However, at this point the investigator finds himself on the brink of an abyss gazing out over completely unknown territory, since no satisfactory method has been developed for measuring the absolute intensity of cavitation, either in the laboratory or in the field.

CAVITATION "SCALE EFFECT"

A common question regarding laboratory studies is, how can the results be extrapolated to the field conditions for larger sizes and powers? If these studies have developed a basic understanding of the mechanics of the phenomenon, the answer is quite definite. Until this goal has been reached it may be very difficult to predict the scale effect. This is the present case for many hydro-dynamic phenomena, including cavitation. It is often assumed that geometrically simlar models and prototypes will have the same cavitation performance when operating at the same cavitation parameter K. Since K involves only pressures and velocities, this is equivalent to assuming that cavitation is independent of size.

However, recent investigations indicate that a change in size does affect cavitation even within the relatively small variations possible with laboratory equipment. Also there is good evidence that the properties of the liquid and its *undissolved* impurities play a significant role in its cavitation performance, but as yet litle has been done to control these properties other than to maintain a relatively constant temperature and *dissolved* gas content.

Laboratory evidences of scale effect have all been consistent, i.e., a suppression of cavitation until after the local pressure has dropped below the vapor pressure. This means that under some conditions liquids have an effective tensile strength. This appears to increase as either size or velocity decreases. From such evidence the author has concluded tentatively that for a given set of liquid conditions the significant parameters are size and shape of the test body and velocity of flow, since these determine the intensity and duration of the tensile stress to which the liquid is subjected before cavitation begins; and that the depression, $\triangle K$, of the incipient cavitation parameter below the value computed on the basis of zero tensile strength is a measure of their combined effects. It is believed that probably some relationship, $f(v, d, \triangle K) = \text{const}$, exists. Such a relationship, if it could be found and verified both experimentally and analytically, would give a direct method of computing scale effect.

NEEDS FOR FUTURE WORK

It is always dangerous to attempt to peer into the future and to make predictions as to what will take place. Therefore the author will content himself with a brief estimate of some of the urgent needs for additional information.

The most important immediate need is for more detailed study of the mechanics of cavitation, in particular that of the "fixed" type cavity. Such a study is inherently involved with the physical properties of liquids and their normal impurities.

Next would come the investigation of the mechanics of cavitation damage. This does not imply secondary importance but reflects the fact that progress here must wait further clarification of the mechanics of the cavitation process.

It is believed that the most serious lacks in our knowledge of the effects of cavitation on performance are a quantitative understanding of the mechanics of the increased resistance to flow produced by cavitation, and the location and extent of various types of cavitation which occur in hydraulic machines, and the effects of changes in operating conditions on these quantities.

Future work on the relative resistance of material to damage should be correlated with studies of the mechanics of damage. An important step would be the development of a definition of intensity of cavitation and some rational measure of it which could be used both in hydraulic machines and structures and in equipment employed for determining relative resistance.

In the study of scale effect, additional carefully controlled experiments would be valuable as the amount of information now available is too meager and limited in scope even to outline the field. In this field scrupulous attention must be given to secure exact geometric similarity if valid results are to be obtained. It is anticipated that the analysis of the results in this field will be greatly assisted by any increase in our understanding of the mechanics of cavitation.

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FUNDAMENTALS OF HYDRODYNAMIC LUBRICATION

ACKNOWLEDGEMENT

This is an abridged version of a paper presented before the First National Symposium of the American Society of Lubrication Engineers. Its author is Mayo D. Hersey of the U. S. Naval Engineering Experiment Station, Annapolis, Maryland.

1. ENOUGH LUBRICANT

Hydrodynamic lubrication means fluid film lubrication, as distinguished from the limiting case of boundary or "adsorbed film" lubrication. The first fundamental is to be sure to have enough lubricant present to separate the surfaces. Just how much that requires would be an interesting problem for calculation and experiment, taking into account surface roughness. Figs. 1 and 2 show some electrical measurements on the oil film in a one-inch-diameter journal bearing at M. I. T. (2, 7).*

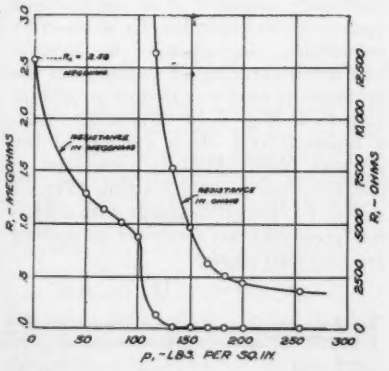

Figure 1. Electrical Resistance and Load (Maximum resistance 2.58 megohms for concentric journal at zero load).

The first figure gives the decrease in resistance with increasing load,

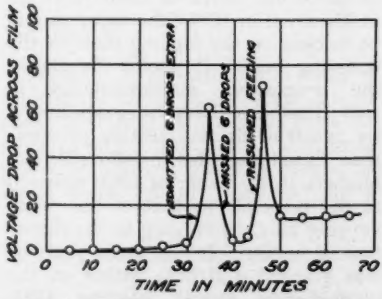

Figure 2. Voltage Drop Across Oil Film (Mean rate of feed, two drops per minute).

*Numbers in parentheses refer to bibliography at the end of the paper.

at a constant feed of 60 drops per minute. The second figure shows the voltage drop across the film between bearing and journal with constant line voltage. The peaks of this diagram reveal the quick response of a starved bearing to a few extra drops of oil.

Even without exact scientific knowledge we can avoid the mistake made in a Connecticut factory, where the line-shaft bearings were installed without oil holes. The story was told by the sales engineer from an oil company whose product was blamed for the hotboxes.

2. NOTHING BUT THE LUBRICANT

A second fundamental aims to have nothing but the lubricant present; avoiding aeration, and abrasive particles, between the two surfaces. The size of grit particles limits the safe film thickness and load capacity. Howarth, describing Kingsbury practice, tabulated what were considered to be the safe minimum film thicknesses at the trailing edge of thrust shoes (31). These values ranged from about ⅓ of a mil, or thousandth of an inch, for the smaller diameters and lower speeds, up to say 3 mils for the larger diameters and higher speeds in current use at the time. McKee, in 1927, had shown that abrasive particles were harmless when smaller in size than the minimum film thickness in journal bearings. Roach experimented at a later date on bearings containing abrasive particles that were large enough to become embedded in the bearing metal, thereby causing journal damage and excessive frictional heat (96). The ideal would be not to allow solid particles to get into the lubricant any bigger than those originally present in the oil drum.

3. THE CONVERGING FILM

A third requirement is that the film be thicker at inlet than at outlet; or to quote the words of Lord Rayleigh at Montreal in 1884, "the layer should be thicker on the ingoing than on the outgoing side." Reynolds developed the proposition mathematically in 1886. Professor Kingsbury applied it by constructing and testing pivoted-shoe thrust bearings in 1898 (33, 76). Michell, in his paper of 1905, referred to the principle as that of the "converging film, converging in the direction of motion." In the same year, he was granted a British patent on the pivoted-shoe thrust bearing (79). Gibbs described thrust bearings with a fixed taper in 1918 (4). Linn and Sheppard described bearings with fixed tapers, both circumferential and radial, at a later date (39). Thus the idea grew up of an "oil wedge" or continuously tapered film. Rayleigh's statement was broader. He showed by calculation, accompanied by experiments with an English penny, that the optimum profile was that of an abrupt step as in Fig. 3 (5, 70), rather than a tapered land. As a variant of the pivoted shoe, Michell described a flexible-shoe journal bearing (Fig. 31 in Ref. 8). Thrust bearings with a flexible grooved plate mounted on springs are also well known.

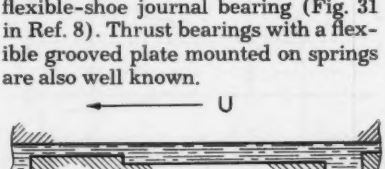

Figure 3. Lord Rayleigh's Bearing (Stepped-film thrust bearing; U denotes velocity of moving surface).

Thus the principle of the converging film requires some kind of geometrical restriction, so as to build up pressure when the lubricant is dragged into the restricted space by the moving surface. The restriction may be an abrupt step or dam, as in Rayleigh's bearing; a flat taper as in many types of thrust bearing; an elastically deformed surface; or the curved, crescent taper of a loaded journal bearing. This fluid pressure supports the load, and is distributed over the stationary surface in such a way that the resultant is equal and opposite to the load on the bearing. The principle has been very simply demonstrated by Boyd (59). His transparent model is sketched in Fig. 4—note especially the tapered recess in bottom of sliding block.

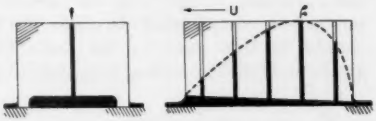

Figure 4. Boyd's Model Showing Film Pressures (Tapered-film under transparent block develops fluid pressure as shown).

4. NO SLIP

Good adhesion is required between fluid and solid. If "slip" should occur, no pressure could be built up in front of the restriction. It is fortunate that ordinary fluids, so far as known, adhere perfectly. Petroff played safe by including a term for external friction in his paper of 1883. This term was dropped by later investigators. The question of adhesion was reopened by Bingham, Deely, and others incident to boundary lubrication. It was also believed for a time that if internal planes of "easy slip" were created by the "regimentation" of polar molecules, so as to reduce friction, it would insure better lubrication. The fallacy of such reasoning will be evident from Boswall's discussion on page 192 of Ref. 15. No kind of slip is wanted, either external or internal, because it would reduce the load capacity of the film. The friction might be low for a moment, but the fluid would soon be gone and you would have abrasion.

5. VISCOSITY REQUIRED

The fluid must be viscous or it will immediately be squeezed out, as had been shown by Stefan some years before the time of Petroff and Reynolds. Stefan was the first to explain what Underwood calls the "squeeze film." Viscosity is necessary to maintain the pressure build-up and prevent contact. Without viscosity, the fluid could not be dragged along by the moving surface and forced against the restriction. Viscosity, however, causes frictional resistance, power loss, and heat generation. It should therefore be no greater than necessary to support the load.

Viscosity is defined as the ratio of the applied shear stress to the rate of shear produced. See Fig. 5. Here an element of the fluid, under a uniform pressure between parallel planes, is being deformed, or "sheared," by a

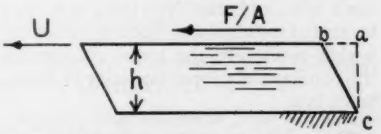

Figure 5. Shearing Under Uniform Pressure (Rate of shear U/h due to shear stress F/A; point a moves to b while c remains fixed).

tangential force F acting on the area A. This action is balanced by other forces brought into play. The ratio F/A is called the "shear stress." Let U denote the velocity of the moving surface and h the thickness of the element chosen. The "rate of shear," sometimes complicated, is given very simply in this case by the velocity gradient, U/h. The viscosity of the fluid is the quotient of F/A divided by U/h.

The part played by viscosity in hydrodynamic theory has been shown by Reynolds and many others. The practical importance of viscosity in reducing wear and oil consumption in automotive applications has been established by investigations like that of Professor Everett (30).

6. NEWTON'S LAW OF VISCOUS FLOW

Ordinary fluid lubricants follow Newton's law, according to which the rate of shear is directly proportional to the shear stress. By means of this law, both fluid friction and film thickness can be calculated in many applications. In the simplest possible application, the lubricant fills the space between two parallel surfaces, and is under a uniform pressure, or no pressure at all. Let the viscosity, assumed uniform, be denoted by Z. Then from the formula given for the definition of viscosity, F/A divided by U/h, the frictional resistance to a uniform motion may be written $F = Z(AU/h)$. As to the units required, if A is taken in square inches, U in inches per second, h in inches and Z in pound-seconds per square inch, F will come out in pounds. Newton's law tells us that Z will be a constant at any fixed temperature and pressure, unaffected by changes in the remaining factors above. Newton's law of viscous flow is an experimental discovery, like Hooke's law and Ohm's law. It is interesting that viscosity may be defined without assuming the truth of Newton's law, and that Newton's law may be stated without mentioning viscosity, which is merely the name chosen for the constant of proportionality in Newton's law.

The above equation for frictional resistance, when applied by appropriate mathematics to a concentric journal bearing, leads immediately to Petroff's law, from which we may calculate the coefficient of friction and power loss of lightly loaded, high speed bearings. An extension to small eccentricities was published at an early date (2). Experiments by Sam McKee and his brother at the National Bureau of Standards provided a correction term for Petroff's law that is still used (17). Friction calculations are more complicated for heavier loads because of the greater eccentricity of the journal, and consequent departure from a uniform velocity gradient, and also because of the greater side leakage. The results of a calculation by Sydney Needs for partial bearings are shown in Fig. 6, taken from (33).

Some lubricants, notably greases, do not follow Newton's law. Other examples are oils below their pour

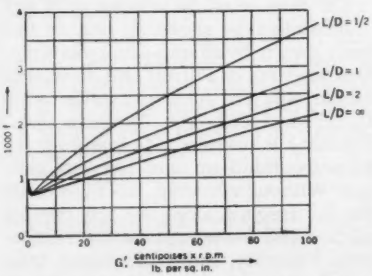

Figure 6. Friction of Partial Bearings (Coefficient for 120-degree bearings from Needs, Refs. 28 and 33, with R/C=1000, and G′ denoting ZN/P in the units shown).

points, and oils that have undergone apparent solidification by very high pressure. The viscosity can no longer be expressed by a single numerical value. The laws governing shear stress and rate of shear in such materials are called "rheological laws," meaning laws of flow. In an ideal, perfectly plastic material there is no shearing deformation, or flow, until the shear stress exceeds a certain value. A live question of the day is to discover the limiting shear stress, if any, above which straight mineral oils may become "non-Newtonian."

7. FILM THICKNESS AND LOAD CAPACITY

The minimum film thickness in tapered film thrust bearings, and in full journal bearings, may be calculated approximately from the theories of Reynolds and Sommerfeld, disregarding side leakage. Load capacity may be defined as the load that will reduce the minimum film thickness to a chosen safe value.

Sommerfeld's solution is well known. It may be represented by the curve in Fig. 7. Here h_0 denotes the minimum film thickness when the radial clearance is c. The ratio h_0/c is called the relative film thickness. The viscosity has been denoted by μ and the remaining symbols are familiar.

Return now to Reynold's solution for the flat, tapered film between rigid, rectangular surfaces; a complicated formula, equivalent to Norton's Eq. [43], Ref. 47. I have taken the trouble to plot numerical values of the relative film thickness over a very wide range of the operating variables, starting out from zero speed. Anyone who does this will be amazed at the changes this curve undergoes.

It starts out concave upward like the Sommerfeld curve, Fig. 7, and bends over, but it has no ceiling or asymptote. The curve may be divided up into

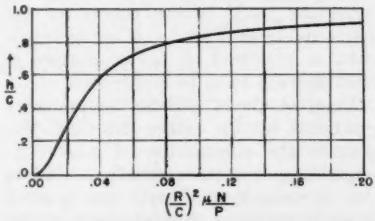

Figure 7. Film Thickness by Sommerfeld's Theory. (For 360-degree bearings without cavitation or side leakage, from Ref. 33; the Sommerfeld number 0.20 corresponding to about 83 cp rpm/psi, R/C =1000.)

intervals, each of which may then be fitted by a simple empirical equation (65). In one of these intervals, usefully located for thrust bearing applications, the film thickness is directly proportional to the square root of the speed, other factors being held constant; see Eq. [24] of the reference cited. At very low speeds, the curve for the tapered land bearing is quite different from that for a pivoted shoe.

8. EFFECT OF TEMPERATURE ON VISCOSITY

The change in viscosity of lubricating oils with temperature has been well described in the A. S. L. E. booklet on "Physical Properties of Lubricants" by Professor Fuller and coauthors. The A. S. T. M. chart on which mineral oils give straight lines was introduced by Geniesse and Delbridge in 1932 (Ref. 21).

As an improvement over the commercially used viscosity index or "VI," Dr. Wilcock proposed a simpler and more scientific viscosity-temperature-coefficient called the "VTC" (50). It is

practically equivalent to a mean value of the ordinary temperature coefficient taken over the full temperature range from 100 to 210° F. The ordinary coefficient is different at different temperatures.

Many of the older viscosity-temperature equations are reviewed in Chapter II of Reference 33. An improved formula was offered by Guy Barr in 1937 (Ref. 35). Another empirical formula, which includes the viscosity-temperature relation as a special case, was proposed by Donald Bradbury (87). This equation is based on new experimental data from 32 to 425° F.

9. TEMPERATURES IN THE OIL FILM

Under hydrodynamic conditions the lubricant itself is the source of the frictional heat. Rapid outflow of lubricant is required if accumulation of heat therein is to be prevented.

Even so there will be temperature gradients set up within the film. The greater the conduction of heat outward from the interior of the film to the surrounding metal, the greater these temperature differences must be. No very practical method has been proposed for determining the internal temperatures of the film, as distinguished from the temperature of the bearing metal. Theoretical estimates thus far made have been reasonably well checked by indirect experiment, notably in the investigations by Kingsbury and by Hagg (26, 49). More elaborate studies have been published by Hummel and by Vogelpohl (12, 69).

Under the simplest operating conditions, the maximum temperature drop across the film at a fixed circumferential position is directly proportional to the product of the viscosity into the square of the speed, and inversely proportional to the thermal conductivity of the lubricant (33).

10. HEAT BALANCE AND TEMPERATURE RISE

Thermal equilibrium and the calculation of steady state temperatures for self-contained bearings were discussed in my paper of 1914 (2) and more fully by Karelitz, with experimental data, in 1942 (46).

The effects of forced lubrication were investigated by Orloff in 1935 (32), McKee in 1940 (44), Boyd and Robertson in 1948 (60), and by Heidebroek and Hagedorn in 1951 (93); also more recently and completely by Wilcock and Rosenblatt (100). A practical study of load, speed, oil flow and operating temperature of roll neck bearings in steel mills was presented before the A.S.L.E. by Hitchcock in 1950 (75). Typical results by McKee are shown in Fig. 8.

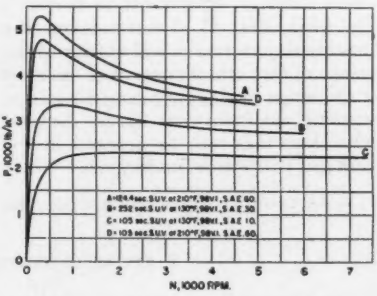

Figure 8. Load Capacity from McKee's Experiments. (Load capacity at first proportional to speed, until viscosity drops owing to frictional heat; finally leveling off with viscosity almost inversely proportional to speed.)

In my papers of 1939 and 1949 (43, 65) the equilibrium problem was reduced to the solution of two simultaneous equations, the first expressing heat-balance, the second representing the viscosity-temperature relation for the lubricating fluid used. The heat balance equation has on its left hand side an expression for the frictional power loss as a function of the mean viscosity in the film. For lightly loaded, high speed bearings such an expression may be derived from Petroff's law. The heat balance equation has two terms on its right hand side, both written as functions of the oil film temperature. One is an expression for the rate of heat transfer (in power units) by conduction, radiation and atmospheric convection; the other for the rate at which heat is carried off in the lubricant (also in power units). In large bearings, with forced lubrication, the latter is by far the dominating term. Thus, two equations are written each containing two unknowns, the mean film viscosity and the mean film temperature. It simplifies the solution if the mean film temperature may be assumed equal to the outlet oil temperature. This assumption was made in Ref. (65) and confirmed by Ref. (100).

Graphical methods of solution with practical applications were offered by Boyd and Robertson, as well as by Wilcock and Rosenblatt. These solutions each involve the plotting of two curves on a viscosity-temperature diagram, one for the heat-balance and one for the viscosity-temperature characteristic of the lubricant used. The intersection of the two curves gives the effective viscosity and equilibrium temperature. Both investigations presupposed a knowledge of the inlet oil pressure, but no knowledge of the oil flow. An analytical solution for any fixed rate of oil flow, without reference to the inlet pressure, was offered during the discussion of Wilcock and Rosenblatt's paper.

In view of the very great drop in viscosity that takes place with increasing temperatures, the importance of the heat balance problem can hardly be overstated. Further study is required on questions like the effect of thermal expansion in altering the clearance, especially during the early stages of operation before thermal equilibrium has been reached. It is also to be hoped that some of the methods already applied to journal bearings may be extended to thrust bearings.

11. HYDROSTATIC AND THERMODYNAMIC LUBRICATION

To initiate hydrodynamic action with minimum wear, motion may be started without load or by "hydrostatic lift." Professor Fuller has modernized and extensively applied the principles of hydrostatic lubrication, supporting the bearing loads continuously by pressure from an outside source (56). Lubrication may also be accomplished by thermodynamic action, as discovered by Fogg (53). The cooler fluid at the entering edge heats up during its passage through the film, developing pressure by its effort to expand. This pressure supports the load, as has been shown mathematically by Cope and others (64).

Few people can remember back to the days when hydrostatic footstep bearings, both oil and water-lubricated, were common practice in vertical hydroelectric generators. Albert Kingsbury became famous for introducing self-contained, automatic oil lubrication by means of his pivoted

shoe thrust bearing (76). This got rid of unreliable pumps that might cause big generators to fail. Yet today we think of hydrostatic bearings and water lubrication as modernistic developments. So indeed they are, in details of application.

Future study may indicate advantageous combinations of hydrostatic, thermodynamic and hydrodynamic lubrication. Possibly, even, the thermodynamic factor may be artificially controlled, instead of being left entirely to frictional heat.

12. THE ZN/P RELATIONS

In the year 1914 Edgar Buckingham of the National Bureau of Standards was writing his first paper on the theory of dimensions (1). We used to lunch together and I wondered if his method would help me out on a lubrication problem. As he struggled with the subject day after day, I began to catch on.

The problem was to publish the results of my M.I.T. thesis with a minmum number of curves, because the data were so volumnous. I tried "dimensions" and it worked. The lucky combination was ZN/P, where Z is the mean viscosity, N the speed in revolutions per unit time and P the load per unit of projected area. Wilson and Barnard (6) were among the first to accept my conclusions.

That the coefficient of friction of a journal bearing could be a function of ZN/P alone, regardless of the separate values of Z, N and P, was doubted by Dr. A. E. Becker (9). He plotted Tower's data of 1883, finding systematic discrepancies. If Becker were entirely correct, this would challenge Reynolds' theory, which was based on Tower's experiments. The discrepancies were sufficiently explained by Barnard in his reply to Becker's paper; but such comparisons are not devoid of interest, even today. Later experiments by McKee at the National Bureau of Standards (14, 17), Hanocq in Belgium (16), and by Boswall and others in England (20) have firmly established the ZN/P relations.

Thus in geometrically similar bearings under normal operating conditions, both the coefficient of friction and the relative film thickness depend primarily on the single variable ZN/P, and are independent of the absolute size of the bearing and of the separate quantities Z, N, and P. It follows that the load capacity, for any fixed relative film thickness, is directly proportional to the product ZN, provided Z is evaluated at the effective temperature and pressure of the film. This relation is consistent with Fig. 8 and may be demonstrated with the McKee lubrication indicator (Fig. 15 of Ref. 33).

With qualified statements that are needed to define the load fluctuations, the same relations can be applied to dynamically loaded bearings. If the surfaces are appreciably deformed, a parameter E/P should be introduced, where E is the modulus of elasticity. With forced lubrication, either p_1/P or D^3N/Q must be held constant along any one curve, where p_1 is the inlet pressure, and Q the flow rate in volume units per unit time. Under more complex operating conditions other "dimensionless variables" have been found that will reduce the number of independent variables.

In any event the data can be plotted on a smaller number of curves, with more points on each curve; and experimental work can be more economically planned, when dimensionless variables are used.

13. AIR AND WATER AS LUBRICANTS

The ZN/P relations indicate that low viscosity fluids like air and water should serve usefully as lubricants in high speed bearings. All that is required is that the fluid be clean; and of course that the speed be high enough, and the load light enough, to bring back ZN/P to the same value needed for lubrication with oil.

Kingsbury's air-lubricated journal is well known. As described in his paper of 1897, it has a diameter of 6 inches with a clearance of less than one-thousandth of an inch, and was tested up to 4400 revolutions per minute. Equilibrium temperature calculations indicate a rise of 90 degrees F in the air film (43). A fortunate property of air as a lubricant is that the viscosity increases with rising temperature.

It has long been customary for people who give lectures to exhibit one of the Kingsbury air-lubricated thrust models, showing how the collar can be spun around while floating on a self-induced, hydrodynamic film of air. Electric contact lights a lamp just before the rotor comes to rest.

Air lubrication on the hydrostatic principle has been described by Mueller and others (94).

Water base lubricants containing at least 98 per cent water have been successfully used in commercial turbogenerator bearings to reduce the fire hazard (37). Complete water lubrication has been used not only for hydroelectric foot-step bearings, but also for non-metallic stern-tube and roll-neck journal bearings (67).

Calculations and model experiments on hydrostatic bearings, including both air and water lubrication, have been published by Gottwald and Vieweg (74).

14. DYNAMICALLY LOADED BEARINGS

Suppose now that you have a rotating load of constant magnitude, rotating at a constant angular velocity: how does that affect the load capacity of the bearing? A shaft carrying a solid, unbalanced mass would exert a centrifugal force of this kind on the supporting bearings.

Orloff in Russia (36) and Swift in England (38) both published a theorem in 1937 offering an answer to that question. The rotating load theorem states that the film thickness and load capacity will be the same as in a steady-load bearing with its shaft running at an equivalent speed of N_e revolutions per unit time. This equivilent speed, it can be shown, is given by the expression $N - 2n$; where N is the speed of the shaft, and n the speed of rotation of the load vector, or line of action of the load. Briefly, N is called the "shaft speed" and n the "load speed." The proof assumes that the clearance space is completely filled with a fluid of uniform viscosity.

From the equation $N_e = N - 2n$ it will be seen that the equivalent speed is equal to the shaft speed if the load remains fixed in direction. It is noteworthy that the equivalent speed drops to zero when the load speed is half the shaft speed. But a steady-load bearing has no load capacity at zero shaft speed. So under these conditions the oil film breaks down, and the rotating load bearing has no load capacity at all, even though its shaft is not stationary.

The theorem was rediscovered ten years later by Stone and Underwood (58) and then, for the first time, ex-

perimentally verified. However, in actual engines the load is far from constant, either in magnitude or angular velocity. Professor Swift, in the reference cited, had set up the differential equations for the most general case, except that his physical assumptions were the same as in Sommerfeld's theory, side leakage and cavitation being disregarded. Professor Burwell and Professor Shaw have attacked the problem where Swift left off (55, 68, 89). Burwell obtained a solution for Diesel engine connecting rod bearings, using the differential analyzer at M. I. T. and pointed out further approximations that may be useful. The mathematical paper by Dyachov (73), and recent experiments by Simons and associates require careful study (73, 81).

We are familiar with the fact that cavitation does occur in the divergent film on the unloaded side of steady-load bearings (63). Yet as I understand Professor Swift and Professor Burwell, they believe that in an engine bearing things happen so fast that there would not be time for the oil film to cavitate. Newkirk was a pioneer in the research work on fluctuating conditions in oil films (29). Professor Newkirk, if I recall his conversation correctly, believes that in an engine bearing, things happen so fast that there would not be time for the cavitated film to close together and fill up.

Is it possible that we are dealing with a somewhat indeterminate case, in which a great deal depends on the accidental initial conditions? In any event it appears that there are two schools of thought, and that the assumptions regarding cavitation require confirmation (92).

15. HIGH PRESSURE VISCOSITY

It is well known that the viscosity of a lubricant is its most important physical property. Other useful properties are the heat capacity per unit volume, thermal conductivity and thermal expansivity. Density taken by itself has little effect except when the lubricant is highly accelerated, or in a turbulent state (82).

Viscosity varies with pressure as well as with temperature. Starting at atmospheric pressure with an SAE 30 oil at 210 F, and applying a pressure of 10,000 pounds per square inch, the viscosity will increase about threefold. This is equivalent to a drop in temperature of 60 degrees F. The relative increase in viscosity would be greater at lower temperatures. Fig. 9, from Ref. 52, shows the relative viscosity of two oils up to pressures of 1000 metric atmospheres (14,200 psi) and above. A medium-viscosity petroleum oil is here compared with castor oil, a high-viscosity fatty oil, at 25 C (77 F) and higher temperatures. See also the recent papers by Blok and

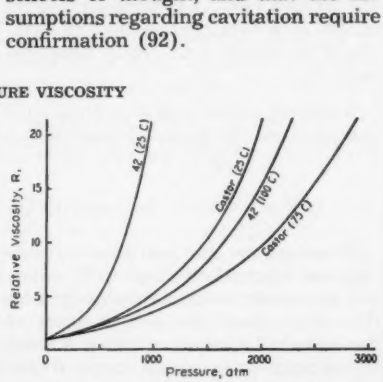

Figure 9. Viscosity Under High Pressure. (The viscosity of castor oil increases 5-fold, that of lighter mineral oil more than 20-fold; both at 25C under 1000 atm or 14200 psi.)

by Bradbury, Mark and Kleinschmidt (86, 87). At extreme pressures the effect becomes so great that many oils appear to solidify. This condition increases the friction, but also improves the load capacity.

Weibull and others have extended Reynolds' theory to include the viscosity-pressure effect (11). Needs

demonstrated the pressure effect by journal bearing experiments (40). He measured the friction torque on partial bearings and found distinctly higher friction under low-speed, heavy-load conditions when lubricants were used that were known to possess high values for the pressure coefficient of viscosity. The results were consistent with the calculations offered and provide a partial explanation for differences in "oiliness" between different lubricants. Needs' data have been analyzed by Blok (84) in his discussion of Ref. 99, using the method of dimensionless variables.

16. ROLLING SURFACES

Figure 10 shows a favorite device used by investigators of roller bearing and gear tooth lubrication. The ratio of rolling to sliding is simulated by assigning any desired values, positive or negative, to the linear velocities U_1 and U_2. Frictional resistance can be measured, but the objective is usually to observe the load capacity of the film as indicated by scuffing, with and without chemically active "extreme pressure" additives.

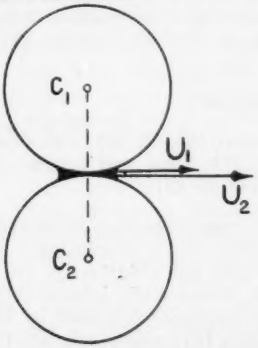

Figure 10. Loaded Rotating Discs. (Steady state shown with center distance C_1C_2 constant, to simulate gear tooth action; for pure rolling, $U_1=U_2$; the condition U_1 = minus U_2 also investigated.)

Hydrodynamic lubrication is the ideal sought for in gear design, but it can be realized only under favorable conditions. Success depends upon excellent workmanship and other factors. Ordinarily the film is partly hydrodynamic and partly of the boundary type.

Hydrodynamic theories of rolling contact lubrication have been developed by a succession of investigators, starting with the simplest conceptions based on Fig. 10 (Ref. 3). Gradually they have tried to take into account the change in viscosity under pressure, and elastic deformation of the surfaces (13, 27, 51, 85, 95, 99). Both factors help to improve the load capacity. In the experimental field, the work of Dietrich, Blok, and Hutt may be considered representative (42, 71, 98), although many other investigations have been published.

Cameron makes an interesting point (91, 99) regarding the customary interpretation of Fig. 10. It has usually been thought that when U_1 and U_2 are equal and opposite, hydrodynamic action would cancel out. The reasoning is similar to that leading up to the Orloff and Swift theorem. It assumes a uniform viscosity. Cameron works out the effect of a temperature gradient across the film. Blok had found that the hydrodynamic film was not completely destroyed in experiments with U_1 = minus U_2. Cameron's theory was offered to explain Blok's observations.

Another limitation to Fig. 10, as a picture of gear tooth action, is the fact that the center distance remains fixed in the diagram. Ought not the centers to be given a motion of approach? Professor Gatcombe has conducted experiments on this question, and will discuss the need for theoretical work.

17. THIN FILM LUBRICATION

It is interesting to think about hydrodynamic action in very thin films, where the roughness of the surfaces has an important influence. This is called "rugulose lubrication" by A. G. M. Michell in his recent book (79).

The term "thin film lubrication" seems to have been first used in one of my earlier articles. It was employed in contrast with "thick film lubrication" to describe the intermediate range of phenomena between classical hydrodynamic lubrication, and the monomolecular type of boundary lubrication. Thin film lubrication includes not only rugulose lubrication, presumably Newtonian; but any kind of long-range boundary film that may exist, like the non-Newtonian films described by Needs (45), and similar films reported by Marcelin (57). The subdivisions of thin film lubrication have been further outlined by Dayton (72).

In rugulose lubrication the heat capacity of the lubricant per unit volume, its temperature and pressure coefficients of viscosity, and the elastic constants of the bearing metal all play an important part. In my paper of 1933 (25) it was shown how these properties might be combined in the form of dimensionless variables. Similar relations were derived by Vogelpohl (34), who found them in agreement with Voitländer's experiments (18). Other experiments in this field have been reported by Cameron, Salama and Barwell (62, 80, 83).

The fundamentals of thin film lubrication have been explored by Professor Blok in his paper before the New York Academy of Sciences (85); and applied to gear lubrication in his discussion of McEwen's paper (99).

18. PRACTICAL CONDITIONS MET

While hydrodynamic principles can be very simply stated one at a time, their application requires critical study and discussion. Consideration must be given to practical conditions like the following:

Alignment of bearings (10, 36, 90); application of lubricant (47, 48, 79); elastic deformation (19, 72, 90); harmful environment (24, 67, 75); and misbehavior of lubricant (30, 48, 67, 77). Also: running-in (14, 48, 83); thermal expansion (33, 68, 79); vibration (12, 29, 54, 77); wear and fatigue of surfaces (48, 68, 83).

Future application of hydrodynamic principles under increasingly severe conditions will require more widespread "lubrication education." This subject has been discussed by Professor Fuller (61) and by the late Professor Norton (47), Professor Bradford, by the author, and others (33, 66, 68). The annual review of literature by Dr. Geniesse will be a valuable aid (97).

19. EXPERIMENTAL TECHNIQUES

Experimental methods must be continually improved to make difficult measurements like film thickness and temperature distribution under hydrodynamic conditions.

Dynamic load experiments have been reported by Cornelius and Barten (41), by Stone and Underwood (58) and by Simons (81), to name only a few. Field research in a steel mill was described by Hitchcock (75).

Film thickness measurements were reported by Needs (45); by Simons and associates (81); and by Wilcock (50, 100). Friction measurements have been made on individual gear teeth by Dietrich (42), using a piezo-electric method; on journal bearings by McKee (14, 17, 44) and by Wilcock (50, 100); and on thin films by Voitländer (18) and by Needs (45).

Temperature measurements of special interest have been described by Karelitz (46), by Hagg (49), and by Roach (96). Vibration techniques have been developed by Hummel (12), by Newkirk and Grobel (29) and by Hagg (54) for investigating oil-film resonance, oil-film whirl and instability in journal bearings.

20. UNLIMITED LIFE

Hydrodynamic lubrication offers unlimited life to bearing surfaces operating under a steady load. It serves to reduce the stress concentrations under fluctuating loads.

A writer in the *Baltimore Evening Sun* (78) predicts 1000 years life for the first Kingsbury thrust bearings that were installed in a hydroelectric power plant. I remember visiting this plant, which is at Holtwood on the Susquehanna River, during the installation period about 1912. Twenty-five years later the bearings were inspected for the first time. The original tool marks on the babbitt surfaces had not yet been rubbed off.

In a lecture by Professor Fuller at Columbia University, which I heard some years ago, the practical importance of hydrodynamic lubrication was brought out by a picturesque statement. He told his students that the best proof of the hydrodynamic theory is to look at the sky line of Manhattan at night with its myriads of electric lights. Every one of those lights—and it must be the same in Chicago—is being generated by machinery that rotates on a fluid film.

Another major application is found in marine propulsion. The oil films in the thrust bearings transmit the entire force that pushes the ship through the water.

21. FUTURE MACHINE DESIGN

Barber and Davenport, working under Professor Bradford at The Pennsylvania State College, once published a paper entitled "Machine Design for Lubrication" (23). The title implies that a more conscious effort may be seen in the future, to design machinery from the beginning in such a way as to facilitate good lubrication.

I should like to think that machine design in the future, including engine bearings and gear teeth, is going to be directed more and more toward improvements based on hydrodynamic theory. The underlying idea was vividly conveyed in a lecture by Sydney Needs, who used the phrase "Designing an Oil Film." The oil film requires engineering design as truly as if it were a wedge of solid metal, like a knife. The film must never be considered a mere accidental puddle. Successful machine design depends on the design of the lubricating film.

Kingsbury led the way, in this upgrading process for machine design, by his paper on "Optimum Conditions" (22). Dahlstrom put hydrodynamic films in roll-neck bearings (24). Linn and Burwell have proposed heavier loads on marine turbine bearings (77). Michell's book (79) aims to promote sound practice by means of correct theory. Boyd and Raimondi have reduced hydrodynamic theory to a convenient form for use in bearing design (88).

Albert Kingsbury once said that his progress in the field of bearings and lubrication research was due in no small degree to his early training as a machinist (76). His first important discovery was that certain differences in friction, previously attributed to different bearing metals, disappeared when sufficient care was taken in fitting the surfaces. His accurate workmanship created hydrodynamic films where none existed before.

Thus if a century of engineering in America proves anything, is it not that scientific and practical knowledge are most fruitful when combined?

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"ENGINEERING COMMUNICATIONS—WHAT'S WRONG?—TOO MANY CHANGE ORDERS SKYROCKET COSTS"

CAPTAIN R. E. W. HARRISON, U. S. NAVAL RESERVE

THE AUTHOR

was born in Manchester, England, in 1893. Educated Manchester College of Technology. With Royal Engineers, British Army 1914-1919, Egypt, Gallipoli, Sinai, France, Belgium. Industrial Engineering experience includes special apprenticeship and supervisory positions with Churchill Machine Tool Company, England, in the building, design and engineering selling of precision grinding machinery; in 1926 as Director of Engineering, Cincinnati Grinders, Cincinnati Milling Machine Company; 1934 Chief, Machinery Division, U. S. Department of Commerce; 1935 Vice-President Chambersburg Engineering Company; in War II, on staff of Assistant Secretary, U. S. Navy, and later as Special Assistant to the Under-Secretary; Registered Mechanical Engineer in Ohio, New York, Pennsylvania, Maryland and District of Columbia. Fellow ASME and Royal Society of Arts; Member other Engineering Societies in USA and Europe. At present engaged in Engineering and Management Consulting, Washington, D. C.

Evolution of a newly designed piece of machinery usually follows a well-established pattern of ten major steps which include functional concept specification, through engineering design specification, working drawings for foundry, forge, machine shop, erection, test and evaluation in the field.

Not infrequently the functional concept is the brain child of a brilliant strategist in the particular field under attack—but a strategist who knows little or nothing about the available techniques whereby his concept must be translated into actuality. Unfortunately for the engineer, it often falls to his lot to work out the mechanical design for realization of a concept which seems doomed to failure because the first line of communication has been blocked, a situation which need never have arisen had the right engineer participated in the spelling out of the concept.

PERFORMANCE SPECIFICATION

Argument to the effect that the best concepts have frequently been arrived at, uncluttered by engineering participation cannot be gainsaid, and this fact is borne witness to by many of our highly successful and highly developed household appliances which have had their genesis in the brain of the harassed housewife, seeking ways and means to lighten the toil and shorten the time of her tasks, to provide that extra couple of hours for a bridge session.

It seems that this first break in the chain of communication between conception and machinery specification is something we will have to live with for a long time; however, it is this author's plea that a competent engineering opinion be conveniently placed, when the strategist feels a good concept is about to be born. While it is true that the engineering mind is essentially conservative in its first approach to a new problem, there are still those amongst us who tremble not at the challenge, and their presence at the birth of that new concept can speed delivery of the finished product. Modern surgery owes much to the improved tools of the trade which have been devised as a result of direct personal team work between surgeon and engineer, and neither could have achieved this progress if denied the skill and knowledge of the other.

So when we want a new ship—an improved cannon—a new locomotive—crane or sewing machine—and the fertile brain puts the imagination to work in devising the desired result, it is wise council which suggests, let's have an unconservative engineer at hand and simultaneously weld in place that first link in the chain of communication.

The new performance specification can take on a more convincing air of authority, if the foregoing procedure be followed, and much time and money in the field of research be saved.

THE ENGINEERING DESIGN SPECIFICATIONS

At this point in the evolutionary process, the coxial cables of communication should really go to work. Actually there are relatively few engineering problems which have not in part been solved before, and the results recorded in an accessible place.

Time spent at this stage of the game in really searching for precept and example will be repaid many times over. Well stocked libraries, information index services, eager trade magazine publishers, and accessory manufacturers, can all prove wells of valuable information tailored to avoid repetition of previous failures and the instigation of redundant researches into sectional problems which have previously been explored and expounded, and sometimes exploded.

Here the lines of communication can really be made to function fast and accurately providing they are energetically prodded.

It is allowed that there is much room for improvement in the speed with which bibliographies on a given subject can be conscienciously explored; some day we will be able to sit in front of an easy to look at television screen while all relevant data is presented in an orderly way for culling that which is relevant. This means, however, a needed improvement in information imparting technique. The information

is there now, but it takes much work and patience to extract and apply it.

To speed up this plan of research into past experiences, many devices have been tried, and probably the best now available are those used in World War II to teach officers and technicians. This work needs expanding—it takes entirely too long today to extract pertinent facts out of these histories, to record and analyze them.

FIRM DESIGN SPECIFICATIONS (BROAD BASE)

At this point extremely sound judgment is required from the directing head of the engineering department, as from here on out mistakes all carry a progressively more expensive dollar tag.

Briefly—that which is mandatory is that the broad base specifications carry the imprint of successful solution (in the engineering sense) to the problem posed by the concept.

Collaterally there must exist the virtue of simplicity to the maximum attainable, the most economical use of materials and man hours, plus the maximum use and reference to existing proven standards applied as subassembly units or components.

It is often difficult for the specification writer to resist the very human tendency to inject his own concept of originality into the specifications, at the expense of using some previously proven standard unit or practice. Only the most rigid control exercised by the finally responsible head of the enterprise can guard against the use of originality injected solely as a sop to the specification writer's own particular ego. Engineers are not by any means immune to this form of wastefulness and only completely adequate two-way communication between Chief Engineer and Specification writer can achieve the necessary economically desirable result.

EXPLORATORY LAYOUTS

This stage is the first real translation of words into drawings. The speed and accuracy with which this phase can be accomplished is largely governed by how good or bad has been the specification writer's work. A skillful and experienced designer, after studying his specifications, can quickly detect the major areas holding design problems, ambiguities in the specifications will quickly be highlighted and liquidated if the channels of communication are kept open—it is equally important that minds be retained in similar status, i.e., "open."

It is at this point that complete attention must be given to the problem of servicing the finished mechanism. Time consumed and customer's costs for service get out of hand if too much reliance be placed on the questionable availability of highly trained service men. Bearing in mind that most user satisfaction has stemmed from those appliances and mechanisms which can be serviced by the user or customer himself, it is automatically apparent that unitization of design is the only acceptable answer. The user replaces units and keeps his screwdrivers and monkey wrenches in his tool crib.

If the mechanism be such that it can

be totally, or preferably hermetically sealed against dirt and the probing screwdrivers and wrenches of the amateur service man, so automatically is achieved an added degree of reliability of continuity of function. This all important feature is fast becoming an absolute must in military equipment of every kind. There simply does not exist either enough time or money wherewith to train technicians sufficiently skillful to entrust with the delicate detailed mechanism which direct gunfire, control electronic communications, and feed fuel to greedy engines.

A simple warning device, either a flashing light or an insistent buzzer, should be the signal for the replacement of a unit. Salvagable units must be serviced at Bases, where the few fully trained, available technicians can be conserved and worked at maximum productive efficiency. Service on the field of battle simply must be speeded up and made more foolproof. If the replaceable unit be hermetically sealed and properly identified, as well as safeguarded against faulty installation, this is the solution.

Communication between experienced field operators and the designer making the exploratory layouts is a must. This is a two-way traffic which, when consummated, spells "success," functionally and from the servicing angle.

FIRM LAYOUTS

Before firming up exploratory layouts, it is of prime importance that the heads of fabricating and test departments be brought in to contribute their quota of knowledge and experience to make doubly sure that fabrication and test can be accomplished in conformity with the most economical techniques and shop practices. This step is frequently best accomplished by the group conference method; however, it should never be omitted and there should be practical unanimity of acquiescence by all participants in the practicability of the design before final layout drawings are released to the detailers.

The injection of shop lore and practice into engineering design before detailing is a basic necessity, and lines of communication in the well run organization must still be kept open for two-way traffic even after working drawings have been issued to the shops.

CONCLUSIONS

The ten progressive steps outlined in the attached diagram are recommended as an effective antidote to the prevailing practice of issuing a multitude of change orders once a project has been initiated to build a certain piece of machinery.

All too frequently, an analysis of change orders shows that they are the children of engineering confusions born of an ineffective system of communication. Experience proves that there is no real time saved at any stage of the game by flinging infirm drawings and incomplete specifications at the shops no matter how well the latter may be organized to stand such shocks.

EFFECTIVE CHANNELS OF COMMUNICATION

(1) Concept of a new performance (Specification)

↕

(2) Preliminary Engineering design (Specification)

↕

(3) Search for precedent and experience (Research)

↕

(4) Firm Engineering Design (Broad basic Specification)

↕

(5) Exploratory layouts and integration of major units (Drawings)

↕

(6) Firm layouts with firm utilization (Drawings and Specifications)

↕

(7) Detailed layouts with detailed unitization (Drawings and Specifications)

↕

(8) Fabrication including assembly and test (Shops)

↕

(9) Performance evaluation (Against concept) (Laboratory)

↕

(10) Field test and evaluation (Field Record)

Thought contributions by broad-gauge engineer

Recorded field experiences with similar mechanisms performance and servicing

Experience contributions by all heads of fabricating shops and service department

The basic problem to be solved is one of engineering management, and this, to be sound, must have its foundations firmly entrenched in accurate scheduling.

To accomplish accurate scheduling of design and production, the job to be done must be spelled out in detail before it hits the actual producers, otherwise their heritage is confusions which grow progressively more acute the nearer they approach assembly and erection.

There are those amongst us who occasionally figure that they profit by these confusions; perhaps they do—temporarily at least. However, it takes but little backtracking through the history of a job to disclose that all concerned could have profited more, had there been fewer change orders brought about by better two-way communications up and down the line from the time when that genius, with the new concept of performance, decided that his brain child merited a chance to prove itself worthy.

"MILITARY NECESSITY VERSUS ECONOMY"

CAPTAIN R. E. W. HARRISON, U. S. NAVAL RESERVE

THE AUTHOR
Refer to page 69

THE PROBLEM

Decision having been taken to operate in the military field in accordance with a military plan which, presumably, satisfies the foreseeable needs of the international situation, it becomes necessary to devise a supporting logistical plan which, while fully meeting the time and quantity requirements of the strictly military, will, at the same time, be achievable with minimum drain on national economic resources.

This as a problem in logistics sounds simple. However, there are two inescapable unknowns which immediately wipe out completely the element of simplicity. First unknown—is the strategic plan of the potential enemy, and second—the extent of the economic burden which must be assumed when we know the enemy plan, probably disclosed in a surprise attack of the Pearl Harbor variety.

Military science, being the resourceful business which it undoubtedly is, promptly produces a blueprint for remedial military action. Print 1 shows what can be done with that which the military now has—Print 2 shows the results obtainable with what the militray would like to have—and Print 3 shows what the military believes it can get in the way of material support tied to a calendar.

Print 1 is the only truly reliable picture—the assembled task of the homebodies is to reconcile Prints 2 and 3 in as short a calendar as is practicable.

Logistical support for a military plan of operation in time of peace or cold war with little or no actual shooting going on offers great temptations to the unwary to take great liberties with that desirable, in fact necessary, feature which, for want of a better name, can be called logistical "elbow room"—i.e., the ability to mobilize resources in such time and quantity and location as will permit the military to merge and implement their Blueprints 2 and 3.

In a democratically operated Republic, where people have a passion for commercial efficiency and an equally passionate hatred of war, there exists, and always will exist, a philosophy predicated on the belief that commercialism can and will provide the answers to all logistical problems. The difficulty which arises in dealing with this philosophy is that it is only half true, inasmuch as it lacks due regard for the element of time, and in war, time is of the essence and the calendar inexorable.

Given time enough—yes—our incomparable commercial resources will solve all our logistical problems of supply, but, unforgettable is the fact that whereas commercial resources buy time by paying dollars, or failing to acquire dollars, the military must buy time with blood.

The foregoing simple fact dispels the comforting complacency engendered by the half truth.

It is an inescapable phenomenon that a resourceful military organization, going through the agonies of a brutal war, will first ask for the minor resources which are needed. However, if it seems at that time more expedient to create those resources, then, rather than wait for a slow moving economy to act, with the object of saving lives no more, no less, the military will create the minor resources which it lacks.

Retention of these resources when peace has been restored is another matter. The military mind which has been harassed while at war possesses a long memory—and this naturally breeds an understandable reluctance to part with the facilities.

It is equally understandable in a country of great commercial ability that there will be strong pressures aimed at divesting the military organizations of every activity of even slightly commercial nature.

ANALYSIS

As is forever the case, the workable answer lies in intelligent compromise. Confining this discussion to logistics, it is assumed that the military blueprints are thoroughly sound and the best which can be devised by professionally competent people.

The problem then can be stated as the devising of a logistical plan with enough "elbow room" in it to permit the military to draw up its three blueprints in such fashion that they can effectively deal with a determined and resourceful aggressor, who, just for the military hell of it, will make his attack on us in a way calculated to give all (including the military) the surprise of their lives.

Analysis of the foregoing reasoning discloses the unknown factor—i.e., "elbow room" which, perhaps more broadly stated, amounts to economic elbow room so devised as to give the military maximum attainable flexibility in selection of action—or again, in simple words, military elbow room.

ELEMENTS OF TIME AND COST

Pure science discloses that most everything starts from a nucleus. Accepting this thought, and it is pretty well mandatory in conformity with reason that it be accepted, then it follows that the more usable nuclei of

skills and physical resources, which we can achieve in peace or cold war, the more commercial or logistical elbow room we have at our disposal when the enemy attack is made.

Harking back briefly to commercial nuclei—it is a notable feature of successful commercial manufacturing operations today, that the great automotive manufacturing corporations make a point of fabricating in their own factories a well thought-out percentage of all their most sensitive components. While it is true that a very large proportion of the forgings, such as crankshafts, camshafts, gear blanks, minor electrical assemblies, and such items as nuts and bolts are bought from outside suppliers, there is retained for manufacture in the home factory just enough of the whole material requirements to enable the company purchasing agent to exercise control over prices and deliveries.

In simple language—the manufacturer deliberately sets up a balance of productive power which achieves an economically sound commercial balance. He knows as hard fact what it costs to produce a certain widget—hence can talk the outside supplier's own jargon when the chips are down and a contract to supply is under debate.

The manufacturer's own nuclei of skills, plus tools, enables him to keep the price and delivery situations in economic balance.

Compare this enviable position with that of the government when it arrives in the market place seeking a supply of widgets. Uncle Sam has no accurate cost figures and no usable potential threat of using his own manufacturing resources. True, Uncle Sam has the three sealed bid system of buying and some fairly shrewd buyers, but the fact remains that with his economic pants around his ankles, he can neither run nor fight, and delays and price defeats are at times spectacular, to say the least. Having no nucleus of skill plus tools of his own, his bought widgets are frequently late in delivery and costly in price.

While a World War is in progress, considerably more than 95% of all military equipments must, of necessity, be procured from commercial sources. At this time the government arsenals and shipyards, while equally busy, are physically incapable of producing more than 5% of what is needed in seven 24-hour days per week.

In addition to this meager but very important contribution to a massive logistical effort, the arsenals and shipyards must undertake special developments which would otherwise interfere with and retard mass production in the mass type production centers. This contribution is of inestimable value, as any experienced production manager will testify.

Quite frequently, in fact, the designs and production techniques employed by the mass type production manufacturers in time of war have been previously worked out in the government arsenals.

Weapon designs and production techniques of the foregoing types have been generated by the skilled government employed nuclei retained for this very purpose. To dispense with these nuclei in peacetime is to court disaster in war. Why? Because of the inexorable time element and the consequent lack of logistical "elbow room." History is replete with the folklore of the fat and lustful people who neglected to keep a sharp manufacturing "sword" wherewith to keep at bay the thin and hungry "have not" nations.

ACKNOWLEDGED—TIME IS OF THE ESSENCE

How much can we afford to pay for "time," or stated another way—what are we prepared to gamble in dollars wherewith to buy time when the stake is our national liberty?

Ideas vary—as they always will do. One thing which can be agreed upon right away, however, is that whatever we pay, the cost will be heavy because a condition, which is our own (i.e., U.S.) interpretation of democracy, makes it that way. The cost of U.S. labor is relatively so high in the international labor market that anything which must be made on any basis other than the mass production concept is inordinately relatively expensive. Peacetime preparedness for war is a time of experimentation and development where all the tools and weapons of war are necessarily tailor made.

This is one of the cost, or price, handicaps which must inevitably be faced if we are to purchase the "time" element, wherewith to preserve national liberty. Those who guide our national destinies are well aware of this high cost factor, and their inevitable task is to so devise our plan of national security that the cost—high as it necessarily must be—is not burdensome to the point where there is question as to its worth.

There is a perfectly human element of dollar acquisition selfishness in every individual — likewise in all groups and parties—hence, the Solomon type of balanced judgment which the Chief Executive must exercise in this struggle for normalcy in the fight between economy and minimum required national preparedness—with sufficient "elbow room" for change and rapid type expansion in the armed forces and their logistical supporters.

NEED FOR SOME DEFINITE BLUEPRINTS

It is customary for the military mind to exercise by planning retaliation in the face of all possible aggressor situations. Today there is but one possible potential aggressor, hence the planning, military and logistical, is narrowed to dealing with the military resources of that one Empire.

Simple is that one phase—yes—but infinitely complicated when account is taken of the multitudinous resources at the disposal of that artful potential enemy—however, difficult the blueprinting and planning of that task may be, it must be faced and a solution devised.

The type of attack which envisages sabotage and militant action from within our own borders is one which could be very difficult to handle, especially if such action be combined with military attack from the air or sea —or both.

At this point it should be well noted by laymen and professionals alike, that traditional U.S. policy has always been predicated on doing our fighting in some distant land or sea, a condition which the strength of the U.S. Navy and Air Force have traditionally made possible. With conditions as they are today, there is probability we will be denied this policy—and the fight in phase one will be in and over our own cities and shores.

It is doubtful if even the most amateur military strategist—or logistical planner—can debate the need for

military and logistical "elbow room" under these predictable circumstances —the alternative being the Maginot Line concept of defense—or to state the case in commercial terms– the Livonia concept of centralized production with maximum production efficiency, but no real "elbow room" wherein to execute an alternative plan of retaliation.

History is replete with the wreckage of the nations who have made the fatal error of underestimating enemy potential. So it would seem wise to evaluate that which the U.S. faces—and blueprint accordingly. While, undoubtedly, much has been done, there exists unfortunately a notable gap between military and logistical blueprints.

CONGRESS

Into this gap there leaps the Congress of these United States, prodded —and rightfully so—by its citizen manufacturers seeking business in a temporarily falling market. Through their trade associations and representatives in Senate and House, these perfectly legitimate seekers of business demand that the logistical supporters of the field forces cease and desist from all pretense of fabricating activity.

SOLUTION

Obviously the gap should be closed and logistical blueprints drawn to fully support military blueprints Nos. 1, 2 and 3.

Congress—sensitive as it always is to the needs of national defense—could then have a logical answer for those who demand intervention.

The Congress of the U.S.—no matter what political party be in power—can be relied upon to give the military the support they need, once convinced that the requested support is based on a logical logistical plan designed to achieve the desired end, without waste.

"Elbow room" wherewith to maneuver to defeat a desperate and resourceful enemy is a demonstrated necessity in both military and logistical fields.

No political consideration can ever override that necessity—it is a safe bet that the U.S. Congress would be the first to support this contention.

So—what do we do? We prepare our blueprints—in other words, our mobilization plans—we check and recheck them to cut out errors, and then, so present the case to Congress that Congress, in good conscience, can make common cause with those who foresee "the day," the need for speed when that day arrives, and above all, the need for maximum "elbow room" for both fighters and logisticians.

BATTLE OF THE BOILERS

ACKNOWLEDGEMENT

This paper, written by Engineer Rear Admiral Scott Hill, was published in the 16 and 23 July 1954 issues of *The Engineer*. Its interest lies in the fact that the author served in ships with Belleville boilers, which boilers, having been adopted by the Admiralty, were subjected to heavy criticism and were subsequently rejected. His contention is that the adverse criticism was not justified, that the boilers were ill-used, and that they received a bad name because the ships in which they were installed suffered from other troubles having nothing to do with the boilers.

In the last years of the nineteenth and the earlier years of the twentieth century—1894 to 1903—the so-called Battle of the Boilers raged with varying intensity in marine engineering circles, in technical papers, and, owing to the vigor of some of the protagonists—notably William Allan, M.P. for Gateshead—in the daily press.

Towards the end of that century it had become increasingly evident that, if warships were to carry more and heavier armor and armament, the most obvious action would be to reduce the great weight of water carried in the cylindrical boilers, with other reductions in the weight and space taken up by the propelling machinery. Inventors of the maritime nations were advocating a complete departure from the old and tried "Scotch" boiler and offering steam producers wherein the water was carried in tubes heated externally by the furnace and capable of standing higher pressures.

The Admiralty studied the problem and the candidates considered for adoption included the Belleville—a large tube boiler—the Yarrow, Thornycroft, Normand, Reed and other small tube boilers, and the Niclausse—which had large tubes with internal small tubes. The Belleville appealed particularly on account of its apparent sturdiness and because it was already in successful use in the Messageries Maritimes. In addition it required no access space at sides or back beyond that taken up by casings, mud drums and firing space. More room was thus available for fire grate area than any other boiler then known could provide, while the elements of which it was mainly composed could be removed through ordinary hatches, making the large openings then usual in the armor deck unnecessary. These advantages naturally appealed to the Director of Naval Construction.

An engineer officer—Edouard Gaudin—a native of Jersey, and easily mistaken for a Frenchman, was sent on a voyage of investigation in one of the Messageries Maritimes ships. His

TABLE—CLASSES OF BATTLESHIPS AND CRUISERS FITTED WITH BELLEVILLE BOILERS

Type	Class	Dates of laying down		Number in class	Number with Belleville boilers	Number of boilers per ship
		First	Last			
Protected cruiser ..	*Powerful*	1894	1894	2	2	48
Cruiser	*Arrogant*	1895	1896	4	4	18
Cruiser	Early *Diadem* ..	1895	1896	4	4	30
Cruiser	Later *Diadem* ...	1896	1897	4	4	30
Cruiser	*High Flyer*	1897	1897	3	2	—
Armoured cruiser .	*Cressy*	1898	1899	6	6	30
Armoured cruiser .	*Drake**	1899	1899	4	4	43
Armoured cruiser .	*Monmouth*	1900	1902	10	7	—
Battleship	*Canopus*	1896	1897	6	6	20
Battleship	*Formidable*	1898	1898	3	3	20
Battleship	*London*	1898	1899	3	3	20
Battleship	*Duncan*	1899	1900	6	6	24
Battleship	*Queen*	1901	1902	2	1	20

* *Good Hope* was another ship in this class.

report, coupled with the fact that S.S. *Laos*, of that line, had given three years' continuous service without important overhaul, determined the Admiralty—Sir John Durston was then the Engineer-in-Chief—to decide on this boiler for all the capital ships then about to be built.

Accordingly, the gunboat *Sharpshooter* had her existing boilers replaced by Belleville boilers and encouraging trials were carried out. Her new fuel consumption was found to be 1.8 lb per indicated horsepower, then quite a good figure. The large cruisers *Powerful* and *Terrible*, each with forty-eight Belleville boilers, were next put in hand, and during the following years several classes of battleships and cruisers were laid down, as shown in the table.

During the same period steam pressures were raised to 350 lb in some ships and to 300 lb in others. Piston speeds were also increased. These changes introduced new difficulties with steam joints and new problems with the white metal bearings. Other changes were also made and, unfortunately, as the direct consequence, too much space was surrendered in certain ships to the Director of Naval Construction, so that access to the auxiliary machinery was cramped, which made repair and attention difficult. Engine-room temperatures became very trying to watch-keepers and, all-in-all, some of these ships became decidedly unpopular with engine-room hands. The larger cruisers were all "four-funnel" ships. This feature carried much prestige. In China especially, a "four-piecey funnel" ship was looked on with great respect. Nor was the four-funnelled ship without its prestige value on the high seas generally.

This very extensive, not to say overwhelming, order for new ships and, particularly, novel boilers, during the late "nineties" distributed in a comparatively short time amongst dockyards and contractors led inevitably to a variety of troubles. It soon became evident that there had been insufficient "briefing" either of the builders, the engineer overseers, or

of the men who were to take charge of the ships in service. Small differences crept in, or were permitted, and unfortunate departures from experienced French practice were made. Today so important a development would undoubtedly be accompanied by a "Belleville Handbook" and instructional courses would be arranged for officers and men. Had this been done the subsequent story might have been very different. But it was not, even though the possibility of trouble was foreseen. Brassey, of 1896, for example, while commending the Admiralty for its courage in going ahead and not waiting for the Merchant Navy to try out water-tube boilers first, at the same time foresaw considerable teething troubles and expressed doubt as to whether Belleville boilers could be forced.

Apart from the boilers the turbine was coming to the fore in 1898, and there were rumors of liquid fuel as Russia was already trying it out in the Belleville-boilered ship *Russia*. So the Admiralty had plenty of calls on its attention.

As the new ships with water-tube boilers came into service several experienced something a good deal worse than mere teething troubles. There were really serious breakdowns. These breakdowns were by no means always due to the boilers. But a good deal of mud stuck to the latter. Conspicuous amongst the failures was the breakdown of *Hermes*, involving her return home after only a year in commission; and there was also the extravagant fuel consumption of *Europa* on passage from Portsmouth to Sydney. It took eighty-eight days, of which fifty-eight were under steam and the remainder coaling; and consumption worked out at 5 lb per indicated horsepower! That high rate of consumption was, in fact, largely a consequence of leaky condensers, leaky joints and a consequent high fuel expenditure in distilling. But it was the reputation of the boilers that chiefly and unjustly suffered.

In 1896 *Powerful* ran successful trials and her sister *Terrible* did well in the Boer War. But both ships experienced troubles due to cramped machinery space and excessive engine-r o o m temperatures. *Powerful* continued in sea-going service until 1912. *Terrible*, having made one trip to China at 11.8 knots in 1902, burning 200 tons of coal a day, made another in 1904 at 12.6 knots, burning only 100 tons a day, indicating more competent handling and understanding, and disproving the idea, then very current, of rapid deterioration of the boilers.

By 1900 there were already a million indicated horsepower of naval machinery dependent on Belleville boilers. But the Admiralty was by then being heavily criticised for its policy. *The Engineer*, for example, gave a great deal of space to the question, and was highly critical of the departure in boiler practice. In July, 1900, it did not hesitate to write of "the failure of a great experiment," and it joined in the appeal for a "Committee of Enquiry." Comment elsewhere after a paper had been read by Sir John Durston before the Institution of Civil Engineers was, "that he had driven the last nail in the coffin of the Belleville boiler." Another engineer said that the Belleville was probably the worst boiler in existence! The boilers of *Powerful* and *Terrible* were retubed at about this time with solid-drawn tubes—a first fitting in later ships. There was an immediate renewed outcry, as these tubes had to be obtained from Sweden. British makers could not produce the right steel. Simultaneously attention was called to the fact that Russia, Holland, Portugal, and Elswick were trying the Yarrow boiler, and U.S.A.—very tentatively—the Babcock.

Sir Wm. Allan, M.P., was conspicuous in his attacks in the House and elsewhere. In September, 1900, the Admiralty yielded to the clamor and appointed a committee under the chairmanship of Admiral Sir Compton Domville to inquire into the question of water-tube boilers for H.M. ships. All the members of the committee were connected with the Merchant Navy or Lloyd's, except Chief Inspector of Machinery, J. A. Smith. Interim reports appeared in 1901 and 1902, and a final one in 1904. The very first report recommended the abandonment of the Belleville boiler except in ships already too far committed, and the fitting of Yarrow or Babcock boilers in future capital ships. The report was unanimous, except for some soft-pedalling by J. A. Smith. It is of interest to note that the committee also recommended the fitting of a mixture of cylindrical and water-tube boilers. This recommendation was adopted to some extent, but was recognized later to be a great mistake.

Despite the recommendations of the first report, Sir Wm. Allan kept up his attacks. So convinced was he of the demerits of Belleville boilers that in 1903, on hearing of proposed trial runs of *Spartiate* and *Europa* to Hong Kong and back, he offered 2 to 1 against the former and 10 to 1 against the latter completing the trials—a bet he would have lost. The final report of the committee, dated June 12th, 1904, was sent to Admiral Domville for signature. By then the "battle" was dying down. He was flying his flag in the Belleville-boilered battleship *Bulwark*. In his covering letter he wrote:—

"My experience with the Belleville boilers on the Mediterranean station has been very favorable to them as steam generators, and it is clear to me that the earlier boilers of this description were badly made and badly used. We have had no serious boiler defect in any of the ships out here, and the fact that two ships are about to be recommissioned with only the ordinary repairs undertaken shows that their life is not so short as I originally supposed. However, the second commission of these ships will be a very good test of the staying power of their boilers."

But breakdowns and delays had made, by that time, too strong a case against the boilers, even though many of the troubles experienced had nothing to do with them. For example, *Hermes* had to come home after one year in commission on the North American station; *Spartiate* broke down badly on her first effort at acceptance trials—entirely due to condenser leakage and bearing troubles—and *Good Hope* never ran for any length of time at her higher revolutions until all her main bearings had been remetalled by Portsmouth Dockyard. The fact was that the ships then building embodied many new departures and advances. But the dog that got the bad name was the Belleville boiler.

Guerilla warfare and sniping against the Belleville boiler continued for many years, but the "battle," in fact, ended soon after the issue of the committee's first report in 1901. The word "Belleville" does not occur in the index to the second volume of THE ENGINEER for 1902, nor does Brassey's annual of 1903 make any comments. There remained the Belleville-boilered ships. Due to their earlier treatment, or to congenital defects, there were a few Belleville ships which remained "lame ducks" at the opening of the first world war, notably *Canopus*, upon which, for a time after the lost battle of Coronel, the defence of the Falkland

Isles solely depended. They had the reputation of driving their engineer officers to breakdown, or even insanity.

Unquestionably the Belleville boiler acquired a bad name far less through its own defects than through its association with defective other machinery. Nor should the effect of the mistaken loyalty of many naval engineers to the Scotch boiler be overlooked. That loyalty was exemplified up to a few years ago, by an elderly retired marine engineer—a regular attendant at the meetings of the Institute of Marine Engineers—who would generally find the opportunity, no matter how apparently irrelevant to the paper under discussion—to get up and say, "There is naething like the Scots boiler!"

THE BELLEVILLE BOILER

So much for the "battle," now for some detail of the subject thereof. What was the design of this boiler that became the subject of a "battle"?

A Belleville boiler was built up of a number of flattened spirals or elements, each composed of straight tubes of large diameter—4½ in.—screwed into malleable cast iron junction boxes at either end, those in front having hand hole doors. These elements rose from a common feed box in front of the boiler, and delivered into a cylindrical steam drum at the top, which was fitted with a system of baffles to prevent priming. Water level was about four rows from the top. Under steam, feed entered the steam drum, descended to the feed box by downcomer pipes—one at either end—through mud drums, which slowed down the flow, and captured any sediment, which could be blown down as needed.

The automatic feed control consisted of a chamber with float, whose movement varied with the water level, and depended for its efficient working on the almost friction-less movement of a spindle in a gland packed with "Belleville" packing—a metallic mixture, then a patent. A rod was attached to a lever from this spindle, and one could tell if water was passing by feeling it, while if pulled down it admitted extra water. Later automatics have improved on the Belleville and need less care and attention. In ships in which the writer served, the care of these automatics—forty-three in *Good Hope*—was the special duty of one mechanic. They were found to be completely satisfactory and reliable. But unfortunately, the Admiralty—probably from some economic motive—issued a packing to its own specification, which led to much trouble. Chief Engineer Gaudin, upon whose report it will be remembered the Admiralty adopted the boiler, refused to use this substitute, and eventually the Admiralty produced either the real thing, or a true equivalent.

The importance of automatic feeding, now realized as essential where the relation between water content, and steam production is small, was impressed on the writer by Gaudin when first reporting to him on joining the *Spartiate*. Having replied in the negative to Gaudin's question, "Do you know anything about Belleville boilers?" the writer was directed, "Go along and pack an automatic gland, then you will know all about Belleville boilers!"

Economizers were fitted, except in the earliest ships, built up of similar, but smaller, elements to those of the

main boilers. They preheated the feed, and were invaluable in Fleet work for controlling the output of steam on sudden stoppages or alterations in speed. By flinging open their doors, draft could be checked, and the discharge of clouds of smoke—so important in coal-burning days—prevented.

A unique device was the junction of the bottom of the element to the feed-box by means of a single "anchor" bolt, with a coned nickel sleeve as the jointing material. This, the most delicate part of the boiler, unfortunately, needed more gentle handling and treatment than was reasonably to be expected from men only accustomed to cylindrical boilers, and the traditional use of a good deal of brute force in their maintenance. Freedom of expansion of the back junction was thus provided for, but it was important that this back box did not get fixed by the accumulation of soot. The disastrous and fatal explosion in *Good Hope* at Gibraltar, soon after her return from the much-publicized trip to the Cape of Joseph Chamberlain in 1904, occurred when the anchor bolts were being tightened under steam. This accident, of course, revived all the old distrust of the Belleville and, in fact, of all water-tube boilers, and probably further delayed their introduction into the Merchant Navy.

A troublesome fitting, later discarded as unnecessary, and almost an inventor's fad, was the provision of fusible plugs in two of the front junction boxes of each element. They were intended to blow if water level reached danger point. But they frequently blew without real cause. In theory they could then be replaced by means of a "pistol." But few became expert in the use of this device and the majority never succeeded in mastering it. Such replacements under steam were seldom reliable. So the plugs were usually riveted over inside when the boiler was cold, or, contrary to regulation, replaced by steel "plugs," leaving the care of the water level to the automatics and watchkeepers. After all, one or more jets of steam at 350 lb pressure, though only through a 3/16 orifice, were found very disconcerting to efficient firing, during the time when someone on the floor plate struggled to stop them with a pistol!

Blowing engines were a part of the outfit, to supply air above the coal surface, but with good stoking their use was often discarded, though they were useful for tube sweeping in harbor.

Feed pumps were of Belleville design, up to and including the *Spartiate* class, but were superseded by Weir's in *Good Hope* class and later ships. Though no doubt satisfactory for the steady steaming of merchant ships, under service conditions they would suddenly and unaccountably stop. It was necessary to keep a heavy copper hammer hanging by each pump, from which a blow on the valve arm would usually restore them to duty! Weir pumps were a very great improvement, though the larger sizes resented having to deal with the light work of harbor steaming, and would not have responded to a copper hammer! They were kept in good behavior by close and frequent attention to the fit of the shuttle, and like the automatics received the constant attention of one mechanic.

The boiler working pressure of 350 lb or 300 lb was reduced to 250 lb by a large valve in the engine-room, which also acted as a separator in case of priming. Another fitting, abandoned almost from the first, was a non-return valve in the down-comer, but the positive circulation later found to be assured proved it unnecessary and, in fact, with sudden changes of steam output, it probably had an effect quite opposed to its intention.

The Committee of Inquiry listed four primary defects in the Belleville boiler:

(a) Defective circulation.
(b) Necessity of an automatic feed.
(c) Feed pressure greater than steam pressure.
(d) False reading of gauge glasses.

Chief Inspector of Machinery J. A. Smith did not agree, but the prestige of the rest of the committee smothered his objections. In the light of modern experience and practice these objections seem quaint, especially *(b)* and *(c)*, which are now common practice. That the circulation was defective was not true. There is no boiler to this day where the circulation is more direct and positive, and in many modern boilers the travel of the steam, or steam and water mixture, is far longer than the 50ft that so startled the committee. The last objection sprang from inherited distrust of anything but hand control, and memory of disasters that had occurred in cylindrical boilers from water shortage.

The committee also commented on the insufficient preliminary instruction given to those in charge. It may, indeed, be taken as a compliment to the engineering branch of the Navy, that it was capable of taking on such a novelty, without special courses or books of instructions without much more trouble than actually occurred.

SERVICE EXPERIENCE

At first it was not understood how greatly the efficiency of water-tube boilers depends on their being worked at a high proportion of their designed output, though service conditions frequently involve the connection of more boilers than are actually required for economic steaming. During the acceptance trials of *Spartiate*, in 1903, the advantage of maximum output per boiler was demonstrated. Somewhere to the west of Land's End, extreme condenser troubles developed, the density at the condensers almost defying the Service hydrometer. The contractors, Messrs. Maudslay, were then in liquidation, and Gaudin had been appointed to act for the receivers, and so was endowed with unusual authority in relation to the dockyard officers. In this emergency they would have jogged home with all boilers connected and in use, and therefore circulation at a minimum or near standstill. He insisted, however, on reducing the number in use to the least required, if forced, for a reasonable speed, with the result that, owing to the rapid circulation, salt was deposited to a depth of nearly an inch on nearly all the tubes, and not only at water gauge level, as would have been the case if steaming easily. What is more no tubes or joints gave out, and all were repairable, though many were badly bent.

Repairs having been completed and final trials been carried out satisfactorily, *Spartiate* commissioned for a "trooping" trip to Hong Kong with relief crews for ships on the China station, in company with the *Europa* similarly engaged. On the return voyage both ships were to carry out a series of trials. The crews embarked were new to the boilers, and at least half the stokers were newly entered "second-class" and wholly inexperienced. But as the crew for the homeward journey would be mainly seasoned men from the Belleville boilered battleship *Ocean*, the trials could be looked forward to with some confidence.

After a fortnight in Hong Kong making good large main steampipe joints—a frequent trouble in those early days of higher steam pressures—the relieved crews embarked, and the two ships sailed for home, having the run to Singapore at economic speed to "shake down."

The trials carried out were:

Three runs of eight hours each at full speed; fifty-four hours at three-fifths power; thirty-two hours at three-fifths power; four-fifths power on the last lap—Gibraltar to Devonport.

The new crew proved at first a great disappointment, as they had been brought up to put no trust in their automatics, but to treat their boilers as if they were cylindrical, and only to be forced at the quarterly trials. However, by the time Singapore was reached, they had been bullied into leaving the automatics alone, and into firing and cleaning the furnaces in the way already proved successful in *Spartiate*. At full power *Spartiate* developed 18,000 i.h.p. and made 20.75 knots, the *Europa* less, but still a creditable speed allowing for the boilers of earlier design, and troubles other than those due to her boilers. All *Spartiate's* trials were completely successful. On the outward trip she had burned 2600 tons of coal for 13 knots, the *Europa* 3600 for 10.75. On the return 10,000 miles *Spartiate's* figure was 4500 and *Europa's* 5600. Up to the date the nearest comparable figure was 4000 tons by the *Blenheim* for the outward run.

The appalling fuel consumption of some of the earlier water-tube boilers was largely due to insufficiently covering the wide furnaces and to neglect of thorough cleaning at the sides—thus reducing the effective grate area. In addition, the prevalence of leaky joints and condensers involved extra distilling. Smokebox doors, too, were often too light and did not close properly, while distrust of the automatics led to priming. Incidentally, it was one of Gaudin's maxims that the proper place of the engineer officer of the watch was far more in the boiler-room than the engine-room, where the E.R.A.s could be quite well left to look after the engines, except on entering or leaving harbor, or at action stations.

The good fuel results of these trials, and *Spartiate's* successful trials generally, were regarded with doubt and even suspicion in some quarters. But, as evidence of the genuineness of her coal economy (which would be appreciated by engineers and others of the coal-burning era) there can be quoted the fact that, arriving at Gibraltar and coaling before her record run home, she "coaled" at a rate that entitled her name to be placed on the board then kept at the end of the Mole, as having beaten all the regularly commissioned ships of the Mediterranean Fleet up to that date—proof that she had not acquired any "plush" or excess coal. Fortunate in her chief engineer, she was also fortunate in her commander, the enthusiastic and original Guy Gaunt, and by the time she reached home the morale and esprit of the ship were of an order only achieved as a rule halfway through a commission.

Her small staff of permanently appointed officers and men again became the nucleus of a new complement of "second-class" stokers and inexperienced ratings to prepare for the annual maneuvers, joining up with the regularly commissioned ships of the Channel Fleet. Not only did *Spartiate* give satisfaction, but on a run to the Canaries, when the order came to proceed independently at full speed, she showed her heels to the more recently built flagship—*Bacchante*—and was promptly detached on an independent mission.

A major defect of all cruisers of that period was the distribution of their coal into a large number of awkwardly placed and small and remote bunkers. In consequence, on any prolonged

passage at half speed or over, an increasing number of hands had to be borrowed from the upper deck to trim coal—this, in spite of the large engine-room complements—600-odd in *Good Hope*. During this run to the Canaries, when "Action" was sounded, and all engine-room hands went below, the seamen emerged from the bunkers to their guns, etc., looking like coal miners, and many physically weary from hard work in the bunkers. Maneuvers over and the ship paid off, Chief Engineer Gaudin and the writer were transferred to *Good Hope*, a ship which with most other Belleville ships, lived out their useful lives, no longer the target of controversy. *Good Hope* was in continuous service until sunk at Coronel.

Subsequent to the commissioning of the *Spartiate* the writer was in charge of stoker firing parties lent to contractors for the acceptance trials of, among others, *Cressy, Hogue* and *King Alfred*, all Belleville ships, and all later credits and assets to the Royal Navy. One could not but observe, however, that while one went through her trials without a hitch, others by equally reputable firms took at least twice the time. The difference in each case was due to defective organization, inexperience or even obstinacy.

CONCLUSION

The Belleville boiler having been banished by the committee, it is to the credit of the Admiralty that it remained determined upon water-tube boilers and carried on with experiments with various types. It adopted the Babcock boiler generally for capital ships and Yarrow or other small-tube boilers for smaller vessels. The Babcock, though heavier and occupying more space than the Belleville, gave good and reliable service through the first world war, but the small-tube boilers gave a good deal of anxiety, mainly as a result of one of the less well considered recommendations of the Committee of Enquiry. It was that all tubes must be straight. This insistence on straight tubes led to the two lower drums being "D" shaped, and not cylindrical, from which arose the alarming and not always easily detected complaint known as "wrapperitis," which, together with leaky condensers, was a constant worry to Admiral Jellicoe in the Grand Fleet between 1914 and 1918.

Gaudin's direct appreciation of essentials, and clear statement of his opinions, regardless of the rank and importance of his hearers is illustrated by the following story. Detailed to explain to the First Lord—A. J. Balfour—the nature of this trouble, he was asked, "Who is responsible for this widespread defect?" and replied at once, "You, sir." Lord Balfour, unruffled as ever, remarked that that was very interesting, but how did it come about? "When you were Prime Minister, sir, you appointed a committee of people who knew nothing of naval boilers to investigate. Amongst their recommendations they said 'all boiler tubes must be straight.' Hence the 'D'-shaped drums and cracks at the junction of the tube plate and wrapper.

During the "Battle of the Boilers," it must be noted that the Admiralty was faced with the consideration of other engineering problems. The steam turbine was developing, and there were problems connected with the efficient burning and possible adoption of oil fuel. In both these matters it took a courageous and ultimately justified lead. Was it not equally courageous in adopting the Belleville boiler? Was that boiler really as defective as it was made out to be?

The writer does not think it was. He wonders, indeed, whether, had some other design been the first to be adopted, it, too, and equally, would have had to carry the odium of much prejudiced comment and the misfortune of much uninformed and unintelligent usage.

NON-FERROUS METALS AND MARINE ENGINEERING

ACKNOWLEDGEMENT

This is an abridged version of the Presidential Address delivered at the Annual General Meeting of the Institute of Metals in London on 27 April 1954. The author is S. F. Dorey, Chief Engineer Surveyor of Lloyd's Register of Shipping. Reprinted from the July 1954 issue of *The Journal of the Institute of Metals.*

I.—HISTORICAL SURVEY

Before considering applications of non-ferrous metals in marine engineering, it might perhaps be as well for me to give a brief outline of the outstanding developments made in this field since the foundation of the Institute of Metals in 1908.

During that time, steam conditions have increased, on an average, from pressures and temperatures of 240 lb./in.^{2} and 400° F. to 625 lb./in.^{2} and 950° F., respectively.[1] Associated with this progress have been the change over from coal to oil fuel and the displacement of the steam engine by the steam turbine for high-powered propulsion purposes. Following the experiments on the *Turbinia* in 1894-96, the steam turbine developed rapidly.[2] The adoption of this type of main propulsion machinery for H.M. Battleship *Dreadnought* in 1906 became a ruling factor in the establishment of the turbine for large ships, and the close of 1907 saw the completion of the steam-turbined liner *Mauretania*. Subsequent developments have greatly increased the efficiencies of these machines, and powers of the order of over 40,000 H.P. are now transmitted by single shafts in large ships.

The oil engine, which has almost displaced the steam reciprocating engine as a prime mover, was at a very early stage of development in 1908, 250 H.P. being the maximum power available, for marine purposes,[3] from engines constructed in this country. The gas engine,[4] which was well established for land purposes, offered opposition to the oil engine in the marine industry, though it attained little more than experimental operation on the gunboat *Rattler*. However, the advance of the oil engine has been remarkable, power outputs per cylinder increasing from 80 H.P. to an existing value of over 1500 H.P., and exceeding 2000 H.P. for engines at present under construction.

The operation of a gas turbine at St. Denis in Paris aroused the interest of

the marine engineer in 1908,[5] and as late as 1910[6] the opinion was expressed that engines of this type might be developed for marine installations. Their high temperatures (although reduced at this time by water injection) and low efficiencies presented such formidable difficulties that the gas turbine has been applied as a marine prime mover only in recent years.

Oxy-acetylene welding[7] was, at this early date, employed only for certain limited hull, boiler, and machinery repairs. Steel and copper were considered as suitable for welding, whereas aluminum was regarded as unsuitable for fusion jointing, soldering being preferred. Controversy existed as to whether the blacksmith or the boilermaker should be employed as the welder, in view of their differing knowledge of the fabrication of metals. The extensive use of fusion welding, in engine and boiler shops and in shipyards for constructional purposes at the present time, appears remarkable in comparison with these early practices.

These few examples, selected from an astonishingly large number of developments made in marine engineering, illustrate the progress achieved during the history of this Institute. This progress has been allied with that of the metallurgical industries, which have provided the essential materials of construction and have developed improved methods of fabrication. With the increased security of operating conditions, the low proportion of failures attributable to the quality of the materials employed is indicative of the progress made and standards maintained by the metallurgist.

To-day economic considerations, in conjunction with limitations of supply, still restrict the uses of non-ferrous metals for engineering and constructional purposes. In consequence, where static or dynamic stresses alone control design and service requirements, non-ferrous metals are rarely included in the materials selected. Since the foregoing remarks impose such strict limitations on non-ferrous metal development and usage, it is obvious that for these metals and alloys to be practical propositions, only markedly enhanced properties, offering manufacturing superiority or resistance to adverse service features, will be of consequence.

The wide range of metals included in the non-ferrous group have many and varied inherent qualities which render them of major industrial importance. It must be admitted that single outstanding features rarely constitute sufficient grounds for the selection of a metal for service, but suitable alloying, in conjunction with fabrication and heat-treatment, has largely provided a solution by enhancing overall qualities. In consequence, these alloys serve a most vital role in ensuring efficient operation and adequate service life of machinery and structures, although the proportion employed in relation to other materials may be relatively small.

Such is the case in marine engineering, and although the uses of non-ferrous metals in this field have been reviewed previously,[8] recent improvements and developments have increased the number and types of alloys in service, in addition to widening their general field of application. A further review is, therefore, not unwarranted, and since manufacturing and operating conditions create the demands for existing and improved alloys, this more logical basis of examination will be adopted. Many of the service problems experienced are of long standing and, while a trend towards minimization has been in progress, satisfactory solutions have not yet been attained. If, therefore, somewhat archaic aspects are referred to, justification is founded on lack of a practical solution.

II.—CORROSION

In the first place, the corrosive nature of marine environments constitutes one of the most arduous service conditions metals are required to withstand in their many applications. The long-term appreciation of this fact is indicative of the formidable nature of the problems presented, in which non-ferrous metals have played an important alleviating role.

At the beginning of this century the corrosion problems associated with marine condensers were of a sufficiently serious character for this Institute to found, in 1910, a Corrosion Research Committee, whose purpose was to investigate the causes leading to the failures encountered. The series of eight reports subsequently published, and the guidance they afforded, were valuable contributions towards the solution of a most serious problem. The marine engineer has derived great benefit as a result of these investigations and their continuation by the British Non-Ferrous Metals Research Association since 1930.

The forms of corrosion observed comprised dezincification, impingement attack, pitting, and galvanic action.[9] The benefits conferred by additions of up to 0.05% arsenic in minimizing dezincification of α-brasses and aluminum brasses are now well established.[10] Since the $(\alpha + \beta)$ 60:40 brass does not respond to such additions, the use of this alloy has been confined to components of working sections which have sufficient thickness to tolerate reasonable amounts of dezincification.

Impingement attack or corrosion erosion was observed to result in rapid deterioration of the alloys then employed. The high local liquid velocities, associated with air bubbles in excess of a critical size, produced a scouring action capable of removing protective scales and deposits, thereby accelerating corrosion. Increase of temperature and any form of local obstruction, which increased liquid velocities, enhanced this effect. Since copper undergoes erosion due to liquid velocity alone, it has now been confined to less arduous services in water systems. The arsenic-bearing 70:30 brass is still employed for condensers in which liquid velocities do not exceed 7 ft./sec. and provides suitable service life in these applications. Where higher velocities are associated with aeration, 70:30 cupro-nickel tubes, with iron contents of 0.4–1.0% and manganese 0.5–1.5%, are widely employed or, alternatively, 76:22:2 aluminum brass inhibited with arsenic is used.

In view of the benefits obtained from small additions of iron and manganese to 70:30 cupro-nickel, the influence of iron on alloys of lower nickel content was investigated.[11] Marked improvements of both impingement-resistance and attack experienced under stagnant conditions were obtained from iron additions of 1–2% to alloys containing 5–10% nickel. Alloys of this type with 5% nickel and 1.05–1.35% iron have prospects of being widely employed for condenser and other purposes. The 10% nickel, 2% iron alloy is, however, preferred in the U.S.A.

Tin-bearing copper alloys with tin contents below 8% have so far proved inferior to these alloys, although when the content is of the order of 12% good resistance to erosion has been observed. Coatings of zinc, tin, nickel, or chromium have been applied to the surface of non-ferrous tubes, but have not been considered worthy of continued use. Duplex tubing, in which different alloys are utilized for bore and external surfaces, have frequently

been considered for specific purposes. In America this form of tube is again receiving consideration, combinations of copper, various brasses, aluminum bronze, or cupro-nickel being supplied with aluminum, Monel, nickel, or steel external contact metals. Chemical and refrigeration plant represent the greatest application of these forms of tubing, but as yet few data are available on the service experience obtained.

Existing normal service conditions of marine condensers are well within the estimated limits of the improved alloys. However, operation involving polluted cooling waters still presents an important problem. Cupro-nickels have been proposed as the most suitable tube materials for this type of service, as both 70:30 and aluminum brasses have proved unreliable. It must be admitted, however, that this proposal does not offer an entirely satisfactory solution, and it is desirable that investigations of deterioration in polluted waters should be continued.

Associated with this problem are those involving severe pitting of these alloys due to marine growths. Bio-fouling tends to be more severe on alloys having enhanced corrosion-resistance. In addition to causing pitting by its own action, the growth tends to trap decaying debris, thereby increasing the attack on the metal surface. Adequate water velocities and careful maintenance assist in reducing these forms of deterioration, but as polluted-water and bio-fouling effects are allied, investigations should serve a dual purpose.

The use of dissimilar metals, whether in contact or in different parts of a continuous system in contact with sea water, inevitably leads to cathodic corrosion. The relative value of the electrode potential for each metal or alloy, immersed in sea water, indicates the liability of marine corrosion occurring in the more anodic metal, when two or more are involved. As a preliminary precaution, therefore, metals and alloys of widely differing potentials should be avoided in these instances, although severe restrictions may thus be imposed on the range of suitable materials available.

Steel-to-aluminum alloy jointing is becoming increasingly frequent with the extension of the usage of aluminum alloys to structural and machinery applications. Local contact at faying surfaces of structures is usually avoided by interposing galvanized strip or non-metallic packing and bolting with surface-protected steel bolts and by the application of zinc chromate paint.[12] In some instances unprotected local steel-to-aluminum contact has resulted in rapid corrosion of the aluminum. In machinery applications, when the necessity arises, zinc coating of the steel is usually adopted as the most feasible method of protection. However, when extruded aluminum alloy hollow sections were employed for cooling grids with brine circulations in refrigeration plant, avoidance of corrosion at steel-bodied shut-off valve connections was not achieved, and these grids have been dispensed with.

Steel riveting of aluminum structures exposed to marine atmospheres is, therefore, undesirable, although in other respects it offers a suitable jointing method. In addition, the liability of steel-cored aluminum overhead-line conductors,[13] in similar atmospheres, to bulging, which is considered essentially of electrolytic origin, also illustrates the difficulties of combining these two alloys. In this instance bitumen coatings were found of little value owing to cracking, hardening, or disintegration, and periodical application of suitable greases provided the best protection.

Copper and nickel alloys are avoided in contact with aluminum and its alloys, and the effects of copper-to-steel contacts are minimized by zinc coating the steel surfaces or by the employment of protection blocks which corrode sacrificialy. Zinc is widely used in this connection, the high-purity form being considered the most effective. However, owing to the ineffective bonding of zinc to steel and to the semi-impervious layer of corrosion product formed on the zinc surface after short-time operation, doubt has been cast on the value of this metal as a protector for marine purposes.[14] Despite the possible validity of these criticisms, however, zinc is still extensively used and must be of some merit. Addition of ½% lithium to the zinc has been suggested[15] as a means of obtaining a less-insulating corrosion product, but no record has so far been observed of this alloy finding use in service.

Iron or steel affords protection to copper alloys and is not only beneficial in reducing the corrosion due to electrolytic action, but also it enhances the corrosion-resistance of aluminum brass.[16] Nickel-base alloys, being more noble, are generally confined to applications with stainless steels or to components where the sections are such that some degree of corrosion may be tolerated.

Protective coatings or blocks of more anodic characteristics therefore form the most common method of limiting these forms of corrosion. In recent years the economic and practical value of cathodic protection of buried pipelines and structures has offered an alternative means of minimizing corrosion.[17] High-purity magnesium anodes have proved most effective and, in conjunction with suitable impressed potentials and current densities, have been extended in use to protect marine pipelines, structures, and the submerged surfaces of ships laid up in reserve. Only limited experience has been obtained with seagoing vessels, as many as 106 magnesium anodes being located on the bilge keel. While such refinements might prove effective in extending docking periods for operational warships, the benefits afforded to merchantmen are doubtful, as these vessels are docked at frequent intervals for many purposes.

A most serious corrosion problem exists in oil-tankers, which carry alternately oil cargo and sea-water ballast[18] in the same tanks. Flat surfaces and other regions which do not drain efficiently undergo rapid corrosion. The combined influence of the oil and the sea water promotes this extreme deterioration, the high oxygen contents of the oils being mainly responsible, although the sulphur content has also been suggested as an additional contributory factor. Steaming out of the tanks aggravates these conditions by loosening adhering scale and increasing the chlorine-ion concentration at the higher temperatures involved. Although cathodic protection, involving the use of an active magnesium or other type of anode, has been applied in limited instances, controversy exists as to the economic and practical advantages obtained.

Aluminum alloys are claimed to have a high resistance to corrosion under these conditions and might offer alternative means of reducing deterioration either as spray coatings, linings, or complete tank units. So far these alloys have not been tried, but future development may involve their use.

The deposition of minute quantities of one metal from solution on to another of more anodic characteristics in water-carrying systems in whose con-

struction dissimilar metals have been used, has been observed[19] to accelerate corrosion failure. Copper transmitted in this form to galvanized steel or aluminum components was the metal mainly concerned, but this example demonstrates the further need of careful material selection for the conditions appertaining.

The high rate of corrosion of metals which can be associated with electrolytic action presented marine engineers with major problems over 100 years ago. For example, the severity of corrosion of iron screwshafts running in sea-water-lubricated brass bushes and in the vicinity of copper sheathing, led to the fitting of brass sleeves or liners to the shaft as a means of protection. At the present time, limitations on the selection of the wide range of metals and alloys available to the engineer are imposed by similar problems. Solutions offered by surface coatings, wastage blocks, or impressed cathodic protection cannot always be employed, or are not entirely efficient. An effective solution appears to be far from obvious, and the engineer can only hope that the metallurgist, in his researches, may develop more efficient methods of passivation.

III.—CAVITATION EROSION

One of the most severe forms of metallic-surface deterioration is associated with damage sustained by cavitation erosion. Under the suction conditions of dynamic fluid flow, the surface of propellers, impellers, and similar hydraulic-machinery components may suffer this form of erosion. The collapse of cavities formed on these surfaces is considered to release quantities of energy in the form of localized forces, similar in nature to water hammer. The magnitude of these forces may be sufficient to bend propeller blades, but their confined character more commonly induces highly localized stresses in the metal, producing surface rupture and sever distortion of the crystal structure. Damaged surfaces tend to increase local turbulence, enhancing cavitation tendencies and extending the depth and area of the eroded zone. The mechanism of failure is not fully understood, and controversy exists as to the part played by corrosion.

Marine propellers experience this form of damage to varying extents, depending on the design, the conditions of operation, and the material of which they are made. Cast iron and cast steel are prone to have local regions of porosity which are easily eroded and give rise to a deeply pitted form of attack. Bronzes, which are of a more resistant nature, exhibit eroded regions having the appearance of areas that have been subjected to intense local sand-blasting. Despite their more resistant nature, however, bronze propellers have been known to decrease in weight during service by as much as 10%, attributable to overall corrosion and erosion losses. Such losses not only impair the propeller's efficiency, but also reduce its inertia. In consequence, the one-node mode of torsional vibration of the machinery dynamic system will be raised, and critical vibrations may be placed out of barred speed ranges or brought in close proximity to service or operational speeds.

Improved propeller designs based on extensive hydrodynamic studies are expected to reduce cavitation conditions, but the higher powers transmitted and the turbulent flow imparted by the stern of the ship will limit the benefits likely to be obtained. The ap-

plication of alloys having the maximum resistance to this form of cavitation erosion has therefore been considered. In order that the relative merits of the available alloys might be ascertained, experimental techniques simulating extreme cavitation conditions on the surfaces of suitable specimens have been adopted.[20–23] Erosion-resistances were based on weight losses in standard exposure times and, whilst some degree of variation of results is inevitable, good guidance is given to the relative resistance of the alloys involved. A reasonable assessment of the erosion-resistance of an alloy is given by the product of the surface Brinell hardness number and the corrosion-fatigue resistance expressed in tons/in.^{2} for 50 million cycles of reverse bending. The more highly resistant alloys have values in excess of 800, and in descending order of merit they include austenitic stainless steels, aluminum bronzes, with or without nickel additions, low-nickel stainless steels, silicon Monel, Monel metal, high-tensile bronze, and Turbadium bronze. Below 800, the normal manganese bronzes, silicon bronzes, phosphor bronzes, gun-metals, cast irons, and aluminum alloys are placed. This does not imply that the manganese bronzes which have given such good service have poor resistance, but they are used purely as a basis of comparison.

Cast iron, which is still widely used as a propeller material, should have its surface skin intact, as this increases its resistance to erosion, as do additions of copper, nickel, and chromium. Soft alloy "stopper" metals employed to plug local casting defects are of negligible value.

Nickel deposits, applied to the surfaces of cast-iron propellers of trawlers,[24–25] have provided increased corrosion- and erosion-resistance in service. The severe corrosion, which occurred at fractured sections, and the high cost of the deposit caused this practice to be abandoned, however. Similar deposits applied to bronze propellers have also proved uneconomical. Spray coatings of zinc, lead, and aluminum have been applied to cast-iron propellers in service.[26] Of these metals, pure aluminum was found to be the most reliable, although respraying of exposed areas became necessary at intervals.

Erosion losses were found to increase with increase of sea-water temperature, up to 50° C., used in laboratory tests, and this feature must be taken into account when comparing the merits of the metals in service. Of the alloys tested showing cavitation-erosion characteristics superior to those generally used, the stainless steels and aluminum bronzes appear the most suitable for manufacture and service purposes. Since uniformity of cast structure is advantageous in providing homogenous properties throughout the propeller mass, the grain refinement obtained from iron and manganese additions to 9–12% aluminum bronzes should be beneficial.[27]

Beeching's experiments[22] clearly indicated that aluminum, nickel, or both, as additions to copper or copper-zinc-base metal, resulted in alloys having low cavitation-erosion-loss characteristics. The latest materials used for large marine propellers are copper-aluminum alloys with nickel and iron additions, resembling that specified in B.S. 1400 A.B.2, and the more recent copper alloy with high aluminum and manganese content. Castings having maximum weights of 25 and 35 tons have been made in these alloys, respectively, and even larger castings are under consideration.

Impellers of pumps are invariably of gun-metal, manganese bronze, aluminum bronze, Monel metal, or stainless

steel. Cast surfaces can frequently be retained, thereby offering superior resistance to corrosion and erosion. The last three materials quoted are usually employed where the conditions are most severe in respect of temperatures, stress, and erosion.

The exceedingly low erosion-resistance of aluminum alloys[21] cannot be neglected if these materials are to be used for hull construction and for small-sized propeller castings. The service experience under conditions of erosion has not yet been sufficient to establish the resistance of these alloys, but, should the experimental evidence be borne out, alternative materials may have to be used for hull sections that experience cavitation.

Sprayed-metal deposits applied for the protection or reclamation of surfaces experiencing cavitation erosion have been found to be of only temporary value, both experimentally[21] and in service. The porosity associated with these deposits is probably responsible, and sound fusion welds represent the best method of repair and surfacing.

Both hydrodynamic and metallurgical researches are leading to a reduction of cavitation damage. Assessment of the factors governing metallic failure by this kind of erosion will undoubtedly lead to an improved understanding of the essential qualities of alloys necessary to resist rupture. In consequence, control in the manufacture and requirements of improved alloys may be established on a more scientific basis.

IV.—FATIGUE

Alternating stresses, arising from the excitation of natural modes of vibration of machinery and component mass-elastic dynamic systems, have resulted in many service failures.[28] Mathematical analysis in conjunction with dynamic stress measurements, obtained during operation of machinery installations, has led to a reduction in failures of this type, either by the avoidance of critical vibrations in the vicinity of operating speeds, or by limiting vibration stresses to values below the fatigue strength of the materials employed.[29]

Alloys having enhanced fatigue strength are advantageous where these conditions are involved, in that they permit higher stress levels to be tolerated. In addition, where the actual magnitudes of the stresses are indeterminable, higher-strength alloys can extend service life by increasing the number of cycles to rupture, and may even provide a complete solution. Such improvements were simply demonstrated by the radical reduction in failures of lead cable sheathing, subjected to vibratory conditions in service, by improving the poor fatigue characteristics of pure lead with small alloying additions of tin, antimony, and cadmium.[30] These alloys are now in general use, although aluminum has also been employed for land purposes, as it provides weight reduction and further improved fatigue strength. However, aluminum-sheathed cables are not used for marine purposes, the cathodic quality of lead being more favorable.

While cold working is readily employed for improving the mechanical properties of non-ferrous alloys, the availability of many alloys capable of acquiring enhanced strength by heat-treatment, has offered marked advantages. In many instances, precipitation-hardening or quenching to obtain structures of the acicular martensitic

type, has had such a marked influence as to render non-ferrous alloys competitive with ferrous materials in many applications. In addition, consideration has been given to the development of strength by the precipitation of ordered phases, but as yet this feature has not been of marked industrial importance.

The low corrosion-fatigue strength of plain carbon and many low-alloy steels, confirmed by reverse-bending salt-spray tests of 50-100 million cycles' duration, at less than 4000 lb./in.^{2} has been responsible for machinery failures.[31] Copper- and nickel-base alloys, in particular, offer excellent corrosion-fatigue characteristics, and manganese, aluminum, and phosphor bronzes, Monel metal, and K Monel metal have all been used in preference to these steels for many shafting applications, including screwshafts of limited size. In addition, these alloys are frequently preferred to the chromium stainless steels because of the tendency of this material to pitting, especially in the presence of graphite grease.

Corrosion-fatigue strengths of many alloys have been determined experimentally for cycles of stress in excess of 50 million, and they provide the engineer with useful design data. However, with the exception of steels, there is a lack of information relating to the influence on the corrosion-fatigue strength,[32–34] of mean stress, stress concentration, and combined-stress conditions.

At elevated temperatures the influence of mean stress in relation to fatigue is of marked importance, owing to the deformation due to creep. At a given elevated temperature, the limitations imposed by fatigue strength and creep strength form individual extremities. A method of computing the limiting strength[35] under their combined influence has been suggested as a basis for design. The reliability of this method, however, still requires confirmation by experiment.

The magnitude of vibratory stresses, occurring when natural frequencies of components are excited by external influences, is dependent on the degree of damping available. While the significance of the damping capacity of material has not yet been fully assessed, in instances where accurate determinations have been made low values have been recorded for many metals and alloys.[36] Internal damping cannot, therefore, be relied on to limit alternating stresses to values below the fatigue strength. This does not necessarily imply that the inherent damping properties of metals are insignificant, for, in similar application, a metal of superior damping characteristics might give longer service life.

V.—USE OF METALS AT ELEVATED TEMPERATURES

The advent of the gas turbine focused attention on the requirements of alloys for service at elevated temperatures. The development of both ferrous and non-ferrous alloys to meet these needs has been indeed remarkable. As operating temperatures even higher than those at present employed would be desirable, the search for superior materials continues. In this country, with the exception of the well-established Nimonic series of alloys, ferritic and austenitic steels have so far proved most suitable for these purposes. In the U.S.A. similar conditions prevail, although more extensive use has been made of cobalt-base alloys and alloys containing high propor-

tions of this metal. Both heat- and creep-resisting steels contain high total contents of nickel, chromium, cobalt, molybdenum, titanium, niobium, tungsten, and manganese, thus demonstrating the importance of non-ferrous metals in these applications. Indeed, the total content of these metals frequently exceeds 45%, and for heat-resisting purposes the higher the temperature, the greater the amount of non-ferrous metals present.

To a large extent, gas-turbine alloys have been based on three primary metals, typical being nickel–chromium–iron and nickel–cobalt–iron alloys and, more recently considered in Canada, nickel–aluminum–molybdenum alloys.[37] Stiffening against creep depends in these alloys on the precipitation of constituents, provided in some instances by small additions of other elements. With the possible limits of these types of alloy being approached, materials based on higher-melting-point metals have been receiving greater attention. Chromium, titanium, and molybdenum are the metals mainly involved and, although the purer forms of these metals are not themselves suitable for high-temperature application, alloys may be developed with more favorable characteristics. One alloy,[38] containing chromium 60, molybdenum 25, and iron 15%, withstood a stress of 9 tons/in.^{2} at 870° C. for more than 1000 hr., indicating the potentialities of alloys of these types.

With extensive research programs in progress throughout the world, embracing both metallic and non-metallic materials, in an endeavor to raise operating temperatures as high as possible, the engineer can only await further developments made by the metallurgist.

The search for high-temperature materials has tended to mask the desirability of determining the operational limits of many familiar alloys used at less-elevated temperatures. Information provided by short-time tests is undesirable, as doubts remain as to the degree of stability of structure and as to the reliability of stress assessment based on these findings, when applied to components of installations constructed for long-duration service.

Copper shortening which occurred in rotor windings of large turbo-alternators[39] aggravated by shift working to cope with peak-hour loads, illustrates the necessity for a knowledge of creep characteristics of materials in service at more moderate temperatures. Copper with additions of up to 0.1% silver to increase the creep strength and reduce the softening tendencies following cold work, or alternatively the introduction of suitable aluminum alloys, has provided a solution to these present problems.

Copper-base alloys used in steam-plant applications have received little attention in long-term investigational work, and suitable pressure–temperature design criteria are based on very limited experimental data. Tests of not less than 10,000 hours' duration are essential on many of these alloys, and the failure of copper alloys in gas-turbine heat-exchangers also demonstrates the lack of knowledge of their scaling qualities in many atmospheres. Metal-to-metal contact is not uncommon at elevated temperatures, and galling has frequently been encountered at steam-valve faces and guides. All steels are susceptible to this form of surface rupture, and the use of Stellite surface deposits has proved the most feasible method of overcoming these difficulties. Silicon Monel metal has good anti-seizing properties at these temperatures, but low ductility and high hardness of the cast 3.75% silicon alloys can lead to thermal cracking with temperature gradients.

Surface deposits of Stellite are widely used for I.C. engine exhaust-valve faces to extend the service life at the high temperatures encountered. Oxy-acetylene flame-welded deposits of 80: 20 nickel–chromium alloy are being increasingly employed for the repair of Diesel-engine valves and provide high resistance to corrosion and cracking.

For steam-turbine blading the ferrous alloys have almost completely replaced non-ferrous alloys in new construction. Stainless iron and nickel–chromium ferritic and austenitic steels are the most commonly employed, replacing existing copper and nickel–copper alloys. Non-ferrous materials are widely used for gland purposes, and the most suitable alloys for the various operating temperatures have been quoted by Brown[40] as being: brass up to 850° F., S Monel (3.5% silicon) below 1300° F., and Nimonic 75 for higher temperatures.

Although the engineer can usually make suitable allowance for the expansion of metals with temperature, the use of dissimilar metals presents many complex problems. The joint application of aluminum and steel to ship construction has been receiving attention, mainly from the point of view of achieving a suitable design to minimize induced stresses.[41]

Temperature gradients, involving thermal conductivity in addition to expansion, are of a more serious nature. Distortion and thermal-fatigue failures are becoming more common with increased operating temperatures. Data provided by metallurgists on mechanical and physical properties of alloys and their variation with temperature have been useful in design, but the factors controlling thermal shock and fatigue failure require greater attention if these are to be avoided in service.

For piston heads of I.C. engines, cast iron has given excellent service and is still preferred by many engineers. Its replacement in heavy-oil engines by forged steel was mainly based on increased soundness and pressure-tightness, although the use of centri-spun 13% chromium cast steel is based partly on its superior heat-resisting properties. While 20% chromium steel is considered superior, as yet this material has not been tried. Aluminum deposits on cast-iron heads have been applied to reduce the heat-absorption characteristics, but opinion differs as to the value of such measures. For heavy slow-speed engines, light-alloy piston heads have not been used, as the saving in reciprocating masses is insufficient to warrant them.

High-speed engines, however, employ cast aluminium-alloy pistons to a great extent, as the reduction in reciprocating masses is of great value. Well-established "Y" alloy and R.R. alloys are used, and reliability has been sufficiently good to merit their continued service.[42]

VI.—APPLICATION OF LIGHT ALLOYS

The advantages of light-weight construction, employing aluminum alloys of suitably high specific strength and good corrosion-resistance, are not so marked in shipbuilding as in other transport industries. In addition, the extensive experience obtained in steel construction and the equipment of yards to handle steel fabrication have militated against the use of light alloys when the latter are not readily adaptable to similar processes. The ex-

tension of light alloys to ship construction has, therefore, been based more on long-term policy, and extension of their application is increasing with design and practical experience.

Scantlings of aluminum structures have been based on somewhat greater values than those given by the strength ratios of the equivalent in steel. Weight-saving, therefore, amounts to approximately 50% of a normal steel structure. The advantages for the present use of light-weight construction have been summed up as: [43]

(1) A reduction in topside weight which increases the stability of the vessel, offering advantages not only in new construction, but in reconversion work.

(2) Greatly increased corrosion-resistance of aluminum alloys, particularly against the detrimental effects of petroleum and its products.

(3) Greater flexibility in the design of vessels operating in conditions which impose limitations on draft, beam, and length.

(4) Reduction in contamination of cargo and ease of cleansing refrigeration and other holds.

(5) For Naval vessels, the non-magnetic characteristics and weight-saving are important, the latter permitting increased armor and armament.

(6) Greater flexibility in hull design, permitting increased speed with the same machinery power or less power for equivalent speeds.

Of these items the first has been the most important in merchant-ship construction, finding application mainly in passenger vessels. Topside weight-saving has been effected by manufacturing masts, superstructures, funnels, davits, lifeboats, and accommodation fittings in aluminum alloys. The S.S. *United States* has incorporated in its construction a total of some 2000 tons of these alloys, and a total weight-saving of 15-20% of the original displacement is claimed as a result of the direct and indirect influence of these materials. [44] With the exception of this ship, such extensive quantities of light alloys are not encountered in the many other merchant ships of composite construction. For these ships the quantities used range up to 250 tons.

For structural purposes aluminum alloys with up to 5½% magnesium are considered most suitable. With this maximum magnesium content and small additions of manganese and chromium, the tendency for precipitation of the β phase has been obviated, and so has the stress-corrosion associated with the presence of this phase. Aluminum-4% magnesium alloys (N.P. 5/6) are used for plate manufacture, as they are more suitable for hot working than those of higher magnesium content. The mechanical properties are generally good and meet the maximum stress requirements of 17 tons/in.^{2} with comparative ease. Further, their good welding characteristics render alloys of this composition superior to those of the heat-treatable type of alloy. There is, therefore, a tendency for this material to be more widely used, and in America alloys of similar composition may in future be employed in preference to the existing 61S-T6 alloy. Details of the light alloys used in ship construction are given in Table I, taken from a paper by Corlett. [44]

Alloy N6 (5% magnesium) is used

for rivets, as well as H10 (1.0% magnesium, 1% silicon), the latter having superior extrusion qualities to the former alloy and being, therefore, more favored. However, it is usually employed in the heat-treated (WP) condition, as is the corresponding American alloy 61-S-T4. These alloys do not work-harden to the same extent as N6, but age-harden to give higher proof stresses. Age-hardening of rivets after quenching is a drawback in shipyard practice, but it was overcome during construction of the S.S. *United States* by adopting the aircraft industry's practice of refrigerated storage. A temperature of −10° F. was employed, the rivets being wrapped in aluminum foil or other insulating material to assist in delaying precipitation-hardening between the time of removal from storage and the actual time of driving.[12]

TABLE I

Aluminum Alloys Used in Ship Construction.[44]

(a) *Percentage Compositions (Nominal)*

	Cu	Mg	Si	Fe	Mn	Cr	Zn	Al
N5/6	..	3.9	..	..	0.8	..	..	Rest
N6	..	5	..	..	0.5	..	..	"
H10	..	0.7	1.0	..	0.4	..	..	"
61S-T (U.S.A.)	0.25	1.0	0.6	..	..	0.25	..	"
Lloyd's re- ⟩*	0.1	5.5	0.60	0.75	1.0	0.5	0.1	"
quirements ⟩†	0.1	1.5	1.30	0.60	1.0	0.5	0.03	"

(b) *Mechanical Properties (Minima)*

(Typical values are given in parentheses)

Alloy	Size or Condition	0.1% Proof Stress tons/in.²	U.T.S. tons/in.²	Elongation, %	
				On 2 in.	On 8 in.
N5/6	3/16−¼ in.	8 (15.5)	17 (20.5)	12(13.8)	(10.1)
	¼−½ in.	8 (12.0)	17 (19.5)	12(22.3)	(15.4)
NE6	Up to 2 in.	8 (8.5)	16 (17.0)	18(25)	..
	Over 2 in.	7 (8)	16 (17.0)	18(25)	..
HE10	W	7 (10.0)	12 (15.5)	18(25)	..
	WP	15 (18.0)	18 (20.0)	10(13)	..
61S-T6	Plate	15.6(17.4)	18.7(20.0)	10(15)	..
	Bar	15.6(17.4)	16.9(20.0)	10(15)	..
Lloyd's requirements	All	8	17	..	10

*Non-heat-treated. †Heat-treated.

Rivets are driven both hot and cold, pneumatic methods being used for N6 rivets of the order of ½ in. dia. (cold) and ¾ in. dia. (hot). The heating of aluminum alloy rivets requires greater attention than is usual in steel-rivet practice. Light alloy rivets are worked in a temperature range of 400°-500° C., higher temperatures being most undesirable.

The high-strength, heat-treated Duralumin and aluminum-magnesium-zinc types of alloy are not employed in marine construction owing to their inferior corrosion-resistance. In the clad condition these alloys are not economic propositions, although they have been employed in some German Naval vessels.

Many small craft have been constructed entirely of light alloys, using normal and stressed-skin techniques. They provide means of gaining general experience of fabrication methods, as illustrated by the recent manufacture of a 72-ft. yacht of light alloy welded construction. For the aluminum alloys to be extensively used in ship construction, efficient welding techniques must be developed. The self-adjusting inert-arc method, as used in the fabrication of this yacht, at present provides the most suitable method of fusion welding. However, as yet, this method is severely limited for general shipyard use by the degree of penetration obtained, the need to maintain a clean and corrosion-free filler wire, and the difficulties of maintaining an inert atmosphere during open-air welding. The high degree of reliability attained in the fusion welding of steels has, therefore, placed aluminum alloys at present in a less favorable position.

Ease of handling gives the light alloys great advantages for hatch covers, crankcase doors, and similar applications. In the engine room, however, apart from these uses and piston castings, light alloys are not extensively employed, as the savings in weight are not sufficiently vital. Where these alloys are employed for crankcase, sump, and cylinder castings, the engines concerned are generally used for other land purposes where these features are advantageous. Some engine builders have even considered offering cast iron as an alternative for marine purposes. In consequence, the cast aluminum alloys containing silicon and magnesium find only limited scope in the machinery of merchant ships.

The ultimate tensile strength of aluminum alloys used in ship construction is still relatively low, 20 tons/in.^{2} being an almost limiting value for normal plates and sections. It must, therefore, be appreciated that the development of alloys having superior strength while retaining the other desirable properties of existing alloys, remains an important consideration.

VII.—USE OF TITANIUM

The remarkable resistance of titanium, in the high-purity form or with small alloy additions, to severe attack by both hot and cold sea water has aroused the interest of the marine engineer. Although this metal has not so far been used for marine purposes, the mechanical properties and corrosion-resistance are of such a high order as to render it very suitable for many applications in this industry. The problems associated with its production and fabrication, in addition to its exceedingly high cost, as discussed

by Dr. Maurice Cook in his recent Autumn Lecture,[45] impose severe limitations on the engineering importance of the metal. In addition, the elevated-temperature properties are disappointing, and until greatly improved alloys are developed, little competition will be offered to existing high-temperature materials.

The resources of this metal are sufficiently great and readily accessible to warrant its consideration for the requirements of engineering industries. If conservation of some less abundant metals becomes necessary, titanium may offer a suitable alternative in many applications. The metallurgist has adequate scope and opportunity to develop this metal, and its future clearly rests with him and his endeavors.

VIII.—BEARING METALS

The maintenance of film-lubrication conditions between metal surfaces in relative movement constitutes the ideal requirements of service. Advances in lubrication theory and practice have been responsible for the efficient operation of machinery by limiting surface contact to a minimum. Load and velocity variations may still result in changes from film to boundary lubrication, and the short-time contacts between the metal surfaces are minimized in their effect by suitable selection of the mating materials. Non-ferrous metals which exhibit a low friction coefficient and a high resistance to local seizure, in relationship to journal materials, perform a most important function in this connection.

While these qualities are of major importance, the bearing metal must also have adequate mechanical strength at the operating temperatures to withstand the applied bearing loads and sufficient ductility to tolerate normal amounts of malalignment. In addition, the rate of wear undergone by shaft and bearing surfaces must be exceedingly low and the corrosion-resistance must be of a high order. A great deal is, therefore, required of a bearing metal, and it is not surprising that no ideal alloy exists and actual selection for particular service represents a compromise or an ingenious combination.

Methods of assessing the relative merits of bearing metals with any degree of reliability have not yet been established, and controversy as to the controlling qualities still exists. The maximum loading per unit area forms a major feature in relation to both the strength of the metal and the degree of wear which may be tolerated. The product of this load and the surface velocity, often quoted as a criterion, leaves much to be desired, as variation within this product of either factor may be too wide. Further, the influence of bearing metals on the critical ZN/P value, expressing change from film to boundary conditions, is not of great significance, as the related function is also dependent on bearing dimensions, clearance, surface conditions, lubrication method, and surface and lubricant characteristics.[46] In effect, gains shown in experiments involving lubrication may be attributed to, or dependent on, the mechanical and not the metallurgical features involved.

In practice, two main types of plain bearings are used, thin- and thick-shelled. For heavy-engineering purposes, where heavy loads, low speeds, and flexibility occur, thick-shelled

bearings are employed. Moreover, in I.C. engine practice, when line boring is preferred, thick-shell practice is obviously involved. The shell thickness vary from 8-12 mm. at a journal diameter of 600 mm. to 2-4 mm. for small-diameter shafts. Although thinner shells may appear to be more favorable for the larger bearing diameters, this has not been confirmed in practice, and war-time economy measures involving reduced shell thicknesses proved unsuccessful.[47]

Marine engines in many instances have shafting of normalized plain-carbon steel with Brinell hardness values between 121 and 173, the lower hardness being more commonly used. White-metal bearings, therefore, are employed of the high-tin Babbitt types, containing 80-93% tin. The large surface areas which may be involved in bonding these bearing metals to copper alloys, cast iron, or cast steel present difficulties in obtaining entirely effective union over the whole surface. In the absence of suitable methods for proving the bond —other than hammer testing—keying of the metal is still widely employed as an added precaution, although it offers little improvement against fatigue failure.

Higher-speed engines, with increased bearing loads, high oil temperatures, shaft journals of heat-treated steel with a Brinell hardness of 300 and line-bored bearings employ thick-shelled lead-bronze, usually tin-backed to steel. The refinement of using top and bottom half-shells of different metals (i.e., white metal on the lightly loaded side and lead-bronze on the highly loaded side) is no longer widely employed. Such refinement was based more on considerations of economy than on bearing theory, and hence the change to all-lead-bronze bearings and the disappearance of this technique.

In practice, failures of thick-shelled bearings are divided evenly between (a) wiping and running of the metals, and (b) cracking or breaking up of the surface. A high percentage of failures of type (a) were due to oil starvation arising from neglect. Cracking of bearing metals by fatigue is exceedingly common, and may be associated with high oil temperature or simply with the inability of the metal to resist the loading conditions.

These failures are invariably associated with the bearing halves supporting the high loads in reciprocating machinery. Bottom halves of main and cross-head bearings and top halves of bottom-end bearings, therefore, are most susceptible to cracking, although bottom halves of bottom ends also suffer a fair proportion of failures in practice.

Alternative bearing metals having increased strength while retaining the other qualities of the high-tin Babbitt metals have not yet been produced, and engineers must, therefore, contend with the failures as best they can. Whether the aluminum-20% tin alloys or the 30:30:40 copper-silver-lead alloy, exhibiting higher fatigue properties, will prove superior to either of those described in thick-shell applications has yet to be demonstrated.

More lightly loaded bearings for line shafting and similar purposes utilize lead-base Babbitt metals. These give excellent service, especially in view of the flexibility of marine-shafting installations and the variation in alignment with loading conditions of the ship.

The realization that greatly increased fatigue strength is a feature of very thin layers of bearing metals has led to the development of thin-shell bearing practice. These bearings are not extensively employed in marine engineering, but are found in some

high-speed main and auxiliary engines. Shafts hardened by surface methods are most commonly used with bearings of this type. White metals and lead-bronze layers cast-bonded to steel shells are both favored, although the latter find greatest usage, with or without lead and indium coatings. The excellent anti-friction properties of lead and indium reduce engine frictional losses, and extended use of these metals in bearing applications is expected.

Top-end bearings are made of white metal, lead-bronze or, more commonly, phosphor-bronze bushes, depending on the type of engine, pin hardness, lubrication, and temperature. Bearing metals of similar type are used in pumps, in addition to other bronzes, while for under-water service an alloy of zinc 30, tin, 68, and copper 2% is frequently preferred.

Sterntube and rudder bearings are salt-water lubricated, being of lignum vitæ and gun-metal mating materials. The sterntube bearing supporting the tailshaft and propeller is of vital importance, as the ship must be docked for examination or repair to be carried out. Wear of the lignum vitæ must be kept to a minimum, and as gun-metal has provided a suitable contact metal in this respect, alternatives have not been greatly considered.

Gun-metals within the range 88:10:2 to 87:8:5 or leaded gun-metals of the 85:5:5:5 type are generally employed in the static or centrispun cast forms for these cylindrical liners. Where facilities are inadequate for casting in the entirety, sections may be cast and joined after shrinkage on to the shaft, by fusion-jointing methods. On completion of machining, the liner may be tested to 30 lb./in.^{2} pressure to ensure soundness before shrinking on to the shaft. Contraction stresses in the longitudinal direction are reduced during shrinking by leaving a region of the bore at the liner centre with clearance and preferentially cooling this section first, after sliding on the shaft. Such regions are pumped up with red lead or similar compound, and the necessary holes, drilled for this purpose, are plugged.

Over a period of 15 months the recorded failures of gun-metal liners were as follows:

Severe Cracking	Severe Porosity	Cracking at Neck Ring	Corrosion—Erosion Scoring	Total
10	2	2	6	20

Penetration of sea water between liner and steel shaft may promote corrosion-fatigue failure of the shaft and certainly electrolytic corrosion will occur. Of the liner failures by cracking, cracks of the order of less than 1 ft. in length were found in all but two instances. Splitting from end to end occurred in one of these two liners and in the other two cracks approaching 3 ft. in length were evident. Local cracking and disintegration in the region of the neck ring could have been due to lack of adequate running clearance, but the full explanation of these failures was not found.

IX.—ELECTRICAL FEATURES OF METALS

The high electrical conductivity of copper has placed it in an almost indispensable position in the manufacture of equipment for the generation, transmission, and distribution of electricity. However, in recent years alu-

minum and its alloys have encroached to an increasing extent upon this field, owing to the need to conserve copper and to the greater specific strength and lower density of these alloys. Electrical-conductor grade aluminum, having a favorable ratio of weight to electrical conductivity, was suitably employed for high-tension transmission lines. More recently alloys of this metal have been used as the windings of transformers, motors, and generators for land purposes. However, copper is still preferred to aluminum alloys for marine purposes, as are lead alloys for the cable sheathing.

The influence of alloy additions has been determined in relation to the reduction in conductivity of pure copper. Silver additions of up to 1% result in only small reductions, but offer enhanced resistance to copper shortening.

High-resistance alloys are for preference solid solutions and Constantan (copper 55, nickel 45%) and Manganin (copper 88, nickel 12%) are widely employed; in addition, nickel-chromium alloys are used for high-temperature operation, with or without aluminum, at compositions within the solid-solution region.

Non-ferrous metals offer a wide range of magnetic characteristics which find extensive usage. Nickel forms an important alloying element with iron as a soft magnetic material, the high permeability peaks, at compositions of the order of 50 and 80% nickel, being the most suitable. Permanent-magnet alloys contain a number of non-ferrous metals, principally cobalt, nickel, aluminum, copper, and tungsten. These alloys contain iron as a major constituent, but there are a number of entirely non-ferrous alloys which exhibit useful ferromagnetic properties, although of only limited industrial importance. However, they are sufficient to demonstrate the potential possibilities of non-ferrous alloys in this capacity and, indeed, the overall electrical field offers one of the most important for non-ferrous metal research and development.

X.—RADIOGRAPHY

Casting and welding inspection, involving radiographic techniques, have been established as unparalleled proving methods. The merits of γ-radiography need little comment, with the exception of improvements afforded by the introduction and application of radioactive isotopes. Iridium 192 as applied to steel thicknesses of ½-2 in. has resulted in high-quality radiographs comparable with those obtained by X-ray methods, particularly at the higher thicknesses. The main disadvantage of this isotope consists of its short half-life period of some 70 days. Cobalt 60, having a half-life of 5.3 years and used on thicknesses of steel up to 6 in., is widely employed for castings and weldments of thicker section.[48] Light alloys are more frequently radiographed by thulium 170, which has a lower equivalent kV. and is, therefore, more adaptable to these metals.

Thickness gauging using thallium 204, strontium 90, and iridium 192, in addition to applying back-scattering methods for the measurement of coating thicknesses or gauging plates and tubes for internal surface corrosion and wastage,[49] illustrate a few of the many applications of non-ferrous metals in radioactive condition.

XI.—FUSION JOINTING AND DEPOSITION OF METALS

The jointing, rectification, and repair of metals by fusion methods now form a major industrial process, and the adaptability of alloys to this type of union has become an almost essential feature. A wide range of methods have been developed for fusion jointing, including many having specific applications. Metallurgical soundness and physical properties equivalent to the parent metal provide the desired object of fusion-welding processes, although when high strength is not a primary feature, other forms of joint involving fused filler metal are available.

Low-strength joints, in which fusion of the parent metal is not attained, have wide application. Soft and hard soldering, brazing, and bronze welding represent the chief such methods. Soft soldering with tin-lead alloys, with or without antimony and silver additions, is applied where temperature and stress conditions are of a low order. Strength investigations on these forms of joint have indicated maximum values for a 50:50 tin-antimony alloy which solidifies over a wide range of temperature.[50] The extensive use of soft soldering for iron, copper, zinc, lead, tin, nickel, and their alloys is sufficiently well known to require little comment.

Brazing, or hard soldering, involving deposits of copper-zinc alloys or copper-zinc-silver alloys (the familiar silver solders), provides alloys of superior strength capable of service at higher temperatures. Gas-torch, furnace, and pot-dipping processes, for jointing copper and its alloys, steels and stainless steels, nickel, silver, and tungsten carbide, form extensive assembly processes. Monypenny[51] has discussed these methods in relation to turbine-blade segment assemblies and lacing wire attachment. The detrimental effects of a tensile stress, initiating cracking in steels during hard soldering,[51, 52] cannot be overlooked, and influences of the higher jointing temperature requirements must be considered in relationship to possible physical and structural changes in the parent steels. Monel metal is not appreciably impaired by brazing, slight reduction in strength only occurring.

The application of these methods to copper steampipe flange attachment, providing many years of efficient service, and the jointing of steel shells to copper alloy tube-plates in heat-exchanger practice, demonstrate their reliability. Bronze welding with filler rods having fusion points exceeding 920° C. imposes limits on the parent metals, of which copper, copper-silicon alloys, cast iron, and steels are the most suitable. Although providing good joints, bronze welding is not widely employed, as methods involving fusion of the parent metal are frequently preferred.

In marine engineering the efficient welding of thick sections is essential, and many processes perfected for thin-gauge material have very restricted application. The degree of perfection attained in jointing steels by electric arc-welding processes has not been approached by any method applied to non-ferrous metals. Less extensive usage may be partly responsible for this fact, although greater complications of oxidization, melting points, thermal conductivity, and solidification characteristics have presented more formidable difficulties in these applications. In all instances, the degree of success obtained in fusion welding is greatly dependent on suitable preparation, fluxing, the development of efficient techniques, the operator's skill, and pre- and post-heating requirements. The quality of

welds cannot be directly attributed to the unsuitability of a particular process without foundation on extensive experience. However, by elimination of some of the sources which could be responsible for error, simplification is achieved. Inert arc-welding processes, therefore, offer marked advantages when applied to many non-ferrous metals. The self-adjusting electrodes and argon atmosphere used in these processes have stimulated increased interest in this technique.

Despite the advances in welding processes, the practice of poured welding or "burning on," first established in the foundry, is still used. Repairs to castings of copper alloys, aluminum alloys, and ferrous materials are most commonly encountered, and with careful procedure and suitable fluxing satisfactory repairs, with metal of similar analysis to the base casting, can be achieved.

Deposits of metals and alloys may be applied by fusion methods for reclamation and repair of worn and damaged surfaces or for corrosion-resisting purposes. Depending on the thickness required, either one of the methods referred to may be used or, alternatively, metal spraying may be adopted. The latter method offers advantages of simplicity, and any metal, except chromium, that can be produced in wire form may be deposited. However, since fusion of the parent metal is rarely achieved and the strength of the deposit is doubtful, reclaimed parts should possess scantlings sufficient to withstand the applied loads before rectification. The building-up of bearing surfaces and worn shaft surfaces has been extensively undertaken by this process with good results.

Surface spraying of zinc or aluminum for corrosion-resisting purposes has been applied to steel, zinc being applied to the hulls of small ships and tank surfaces.[53] The tendency for coatings to be porous requires that deposited metals should be anodic to the base metal, and fusion welding is preferable where severe erosion conditions prevail.

Electrodeposition of non-ferrous metal for reclamation, corrosion protection, and decorative purposes has become so well established as to require little comment. Nickel and chromium deposits have proved reliable for the reclamation of worn surfaces and in some cases have been preferred. Of the two metals, nickel has been most favored, and for steel shafting continuous deposits in the form of bands obviate any possibility of the metal flaking off the surface.

Cadmium plating of steel in contact with aluminum has been proposed in constructions involving these two metals. The favorable electrode potential of cadmium could be utilized on the surfaces of steel bolts and similar items in contact with aluminum. Cadmium may, therefore, be used in this form to an increasing extent, engine builders in other industries having already proved its merits in reducing these dissimilar-metal problems.

Non-ferrous metals are readily used in all these deposition processes, but fusion-welding methods have not yet attained the desired efficiency required for marine purposes, particularly in regard to aluminum alloys, some alloys with aluminum contents, and many high-temperature materials. The use of these materials is being influenced by this feature and can be increased only by the introduction and development of more efficient and reliable welding methods.

XII.—FABRICATION FEATURES

A high proportion of non-ferrous metals is used in the cast condition for marine-engineering purposes, copper alloys being the principal materials concerned. A large number of these castings having weights less than 10 cwt. are used for valve bodies and components, pump parts, bushes, brasses, and similar items, while sterntube parts, tailshaft liners, larger pump casings, impellers, and turbine-nozzle castings more commonly weigh between ½ and 5 tons. Propeller castings are frequently of 30-40 tons weight in the cast condition, of which some 25-35% may be removed in machining operations. These items comprise some of the largest castings made in non-ferrous alloys and demonstrate the castability of these alloys and the high degree of foundry control employed. Although castings of 50 tons weight represent the largest produced for marine purposes in manganese-bronzes, the size limitations are imposed by machining facilities and design requirements.

Propeller castings of high-tensile ($\alpha + \beta$) brasses or manganese bronzes are still extensively used, although aluminum bronzes are increasing in popularity as a result of their superior erosion qualities and increased strength, which permit greater flexibility in design. Although modified β-brasses of the high-tensile type offer excellent strength qualities, the liability of these alloys to intercrystalline cracking in sea water forms an undesirable feature. However, a limited number of propellers has been cast in alloys of this structure containing 2% nickel and having tensile strengths of 35 tons/in.^{2}. Nickel additions of this order to normal manganese bronzes increase the erosion-resistance, but tend to increase difficulties of repair and rectification. Turbadium bronze, an alloy of this type, has been employed fairly extensively, the propellers of the *Queen Mary* of some 32 tons finished weight being typical examples.

Few propellers complete their service life without some repairs being necessary to eroded areas or to make good damage sustained by striking submerged objects. Failures due to defective castings are comparatively infrequent, indicating the generally high quality obtained.

Improvements in foundry control, casting methods, and inspection techniques have enhanced the soundness and uniformity of properties of all castings. Although welding methods have provided simple means of reclaiming and repairing defective castings, it is essential that this fact should not be reflected in relaxation of control or of efforts to improve the casting procedure and so reduce the defects encountered.

The improvement in pressure-tightness [54] of bronze castings effected by lead additions has resulted in a preference for this form of alloy. Low-pressure valve bodies and parts, sea connections, liners, and similar parts have, therefore, been cast in both these materials.

Manganese, nickel, and aluminum bronzes have been extensively employed for higher-pressure parts and more severe temperature conditions, although for steam chests steels are being more extensively used, as the upper operating limits of these non-ferrous alloys are being exceeded or have not been sufficiently well established to warrant their use.

Centrifugal-casting methods are extensively employed where the form of

the components is adaptable to the process. Both chill and sand molds are used, the former for gear blanks and bushes, where the improved structures at the chilled surfaces are of importance; sand molds are employed for tailshaft liners and stainless-steel piston crowns.

The higher-strength alloys are taking precedence for cast machinery parts, the aluminum bronzes being the most popular. Difficulties due to the aluminum content do not appear to be unduly severe for casting or welding, provided that reasonable precautions are taken. Vulcan hydraulic coupling rotor and clutch parts have been cast in these alloys, the intricate oil passages being readily reproduced and the strength being adequate for rotational speeds in excess of 750 r.p.m.

Aluminum alloy castings are most commonly used for lower-strength parts where ease of handling is essential. Sand- and die-cast pistons of these alloys are widely used for the smaller oil engines, as in other industries. Die-casting and precision investment-casting of non-ferrous alloys have not been extensively used in marine engineering, although high machining costs have led to consideration of these methods for low-stressed turbine and other machinery parts, at present machined from wrought materials.

Wrought alloys are used for shafting purposes in extruded, rolled, and, less frequently, forged form. The convenience of rolled or extruded round bar is useful for shafting having loose steel couplings. In consequence, as a result of the generally available sections and economic considerations, shaft diameters in excess of 5 in. are rarely used.

Aluminum alloys used in ship construction are not made in such convenient sizes and sections as steels, and a tendency to increase the sizes of plates, in particular, has been in progress to meet these demands. Cold working of plates and sections to obtain more favorable mechanical properties imposes limitations on welding, in that the weld and fusion zone will be inferior in strength to the remaining parent metal.

The use of the plasticity of non-ferrous metals is of value for jointing purposes. Metal gaskets are most commonly of lead, aluminum, tin, nickel, Monel metal, and copper. Selection of the metal depends on the pressure, temperature, type of joint, and possibility of galvanic action with the flange material.

XIII.—CONCLUSIONS

Within the scope of this survey some of the merits and limitations of non-ferrous metals in marine engineering have been considered. Combating corrosion still provides the most important function of these metals, and although improved alloys have reduced the rate of deterioration against some forms of this attack, there is still plenty of scope for further developments.

For elevated-temperature applications, low- and high-alloy steels have replaced non-ferrous metals in steam plant and in steam-turbine and gas-turbine compressor components. The Nimonic alloys, however, are preferred by many manufacturers for gas-turbine blading and similar stressed components.

Light alloys are still at an early stage

of development for shipbuilding purposes. Their future progress is largely dependent on a reduction in cost for the equivalent steel structure, and this can be related to the necessities of improvements in both material and fabrication methods.

While there is a trend for plain bearings of steam turbines to be of the steel-backed, thin-shell, white-metal types in place of the existing thick-shelled, bronze-backed types, similar trends are impossible in heavy-oil engines. Fatigue failures of the thick-shelled, white-metal bearings used in these engines are not infrequent, and bearing metals of improved strength and capable of use with plain carbon steel journals would be advantageous.

Many of the non-ferrous metals used have limitations of resources and production. In consequence it is undesirable that either the engineer or the metallurgist should restrict his views too greatly in considering alloys for particular service. With the large number of existing alloys and the prospect of further increases in the future, close co-operation between engineer and metallurgist is desirable in order that the most efficient and economical selections of material may be made.

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WILL NUCLEAR FUELS RUN MERCHANT SHIPS?

HOLMES F. CROUCH

THE AUTHOR

is a graduate of the U. S. Coast Guard Academy in 1940. He spent five years on Coast Guard vessels at sea. His broad experience includes five years inspection of merchant ships (hulls, boiler, lifesaving and firefighting equipment); two years public relations in Washington, D. C.; two years with a prime contractor to the Atomic Energy Commission (nuclear reactors for central-station power). He was recently with an industrial sales promotion agency in San Francisco and is now pursuing a graduate program at the University of California in Nuclear Engineering.

There has been much speculation and prophesy about the application of "atomic power" to merchant ships. The generalization seems to be that with the advent of nuclear energy, its utilization by large ocean liners is automatic—given time and the proper economic incentives. Evidence of this thinking stems from the many references to the *Nautilus* and *Sea Wolf* as examples of the future attainment of this goal. But, actually, so very little specific and comparative information has been given out that the realistic question arises: Will nuclear power really benefit merchant ships? Everyone says it will, but will it?

Indications are that the two nuclear submarines will prove operationally successful. In fact, publicized long-range plans include the building of two more—making a total of four atomic subs. Obviously, since nuclear reactors producing propulsion power are feasible for ships of the Navy, why not so for merchant ships? Such interest often leads to thinking in terms of a direct substitution of the submarine reactors for conventional merchant marine boilers. However, this is impractical . . . for technical and economic reasons which follow.

SUB REACTORS UNSUITABLE FOR MERCHANT SHIPS

Freedom from dependence on air for fuel combustion makes nuclear power predominantly advantageous to a submarine. The unprecedented tactical "plus" that results from almost unlimited military operation underseas, alone justifies the high cost and complexities involved. In sharp contrast, however, operating without combustion air is only a negligible consideration for merchant ships.

The submarine reactors are of the

"once-through" type. That is, each charge of nuclear fuel is used solely for the production of power. Because of nuclear peculiarities, this is not the most efficient reactor design. A better design would be to utilize those neutrons which unavoidably "escape" from the reactor core, for the breeding of new fissionable fuel. While military considerations assign no premium to the conservation or build-up of vital nuclear fuel while it is being burned, there would be such a premium on merchant ships. This would stimulate "dual-purpose" type reactor systems wherein new fuel would be bred simultaneously with the development of propulsion power. Though these systems would result in larger reactor plants, the increased space requirements would be of no major consequence on merchant ships.

The earlier submarine reactor is designated as STR—"T" for thermal. This means that the neutrons which trigger new fission events have been slowed down—or moderated—to thermal energies. By moderating the neutrons, their probability of interaction with fissionable nuclei of the uranium fuel is substantially increased. Ordinary water (highly purified) is used both as the neutron moderator and the reactor coolant. To maintain the moderating process and simultaneously attain the high coolant velocity necessary for efficient heat transfer from the reactor core, a relatively low water-temperature rise results. The net effect is that the STR produces only *saturated* steam—at 250 psi, 415°F. Steam in this condition would never be acceptable on modern merchant ships where the boiler and turbine technology is too well advanced, and where systems are standardized in the neighborhood of 650 psi, 900°F.

The SIR ("I" for intermediate) submarine is not so limited in steam conditions. Because intermediate-energy neutrons fission new events, liquid sodium is used as moderator and coolant. The boiling point of sodium is 1600°F at atmosphere. Hence, its temperature rise in the reactor core can be allowed to go higher than that of water, to subsequently produce superheated steam—at 500 psi, 750°F. Although this steam is still outmoded by present-day practice, there are other disadvantages of SIR for merchant ships. Sodium burns in air and reacts violently with water, thus requiring extra provisions for handling and leak prevention. Also, radioactive sodium is more hazardous, and its decay is considerably longer than radioactive water. This means that the use of sodium as a moderator-coolant imposes long waiting periods before maintenance and repairs can be made.

LARGEST-FASTEST SHIPS ARE FIRST CANDIDATE

Although there are obvious differences between military and commercial requirements for vessels, the submarine reactors do help us to establish a starting-thought on the type of merchant ships to be considered. It has been concluded that submarines and destroyers are about the smallest practicably size for mobile nuclear power. If we discard tonnage comparisons between naval and merchant vessels, we can accept the submarine propulsion load as the general index of technical feasibility. Submarines of the nuclear powered class are known to develop in the neighborhood of 25,000 shaft horsepower. This is more shp than most merchant ships command. Thus, we need think only in terms of extra-large and extra-fast ships—the "Mariners," Supertankers and newest Passenger Liners. Some general characteristics of these ships are tabulated as follows:

Name	Trade	Displacement	Horsepower	Speed
Passenger Vessels		(T)	(shp)	(K)
America	Trans-Atlantic	35,000	37,500	22
Independence	Mediterranean	30,000	37,000	22
Lurline	Hawaiian	26,000	25,000	20
Super Tankers				
Delaware Sun	Worldwide	39,000	14,850	18
P.C. Spencer	Coastwise	32,500	13,750	18
Large Freighters				
"Mariner" Class	Worldwide	22,500	17,500	20
Metapan	Coastwise	16,000	13,000	16

WHAT ABOUT SAFETY AND EFFICIENCY WITH NUCLEAR POWER?

Regardless of the class of vessel selected—freighter, tanker or passenger—absolute reactor safety would be a paramount design consideration. The problem would be distinctly different from that of nuclear plants ashore producing electrical power. These central utility plants can be located remote from populated areas so that in the event a reactor did "runaway," its release of radioactive fission products would entail no hazard beyond the immediate area of the plant. To prove this, a "Borax" experiment at the National Reactor Testing Station, Idaho, pulled all stops to let a reactor actually runaway. Although the reactor core did melt down, the official report states that: "There was a large instantaneous release of energy accompanied by some radioactivity. There was no overexposure of personnel to radioactivity and no hazards resulted outside the immediate vicinity of the experiment."

But what happens if a nuclear powered merchant ship collides in a fog-bound harbor, or founders in a storm at sea, or catches on fire? Passengers and crew would find it difficult to escape from the "immediate vicinity." The release of radioactivity to seawater would not only contaminate survivors, but would be spread by tides and currents to nearby beach areas. These would be radiological safety problems in extreme.

The at-sea factor of a merchant ship and its power load in port are of major consideration in the operating reliability and the economy of a reactor system. Although a reactor can be started up and shut down any number of times, it is most uneconomic to do so. When shut down, fission products decay to generate new heat which must be dissipated; "poisons" are built up which must be removed—before the reactor can be started again. This means that with longer periods at sea and greater auxiliary loads in port, fewer reactor power changes would have to be made . . . with consequent more efficient plant operation. From these considerations, nuclear powered passenger ships and tankers appear to be ideal. Both are noted for extensive times at sea, quick turn-arounds and significant power demands in port. Freight vessels, because they spend so much time loading and discharging cargo, probably would not be among the first candidates for nuclear power.

FIXED NUCLEAR FUEL IS THE PRIME ADVANTAGE

Nuclear power offers large, fast passenger ships one supreme advantage. The amount of fuel required is fixed: the power available is *independent* of speed, cargo and distance of haul. Here's why.

Any nuclear reactor requires a certain minimum quantity of fuel, known as critical mass, for start-up. This critical amount is large when compared to the minute quantities for start-up of oil-fired boilers. Beyond criticality needs, however, only a small proportion of additional fuel feed is required to keep the reactor going. This "excess reactivity" (as it is called) is less than 10% of the original amount. All fuel inventory is built in the reactor system at one time. Whatever its form (solid or molten), once the reactor is in operation no further fuel external to the system can be added until the reactor is completely shut down.

For a reactor of the power level necessary to propel a ship, the total fuel inventory would probably be "around" *15 tons!* Those who have been conditioned by science-fiction stories of "atomic pills," should refer to the various articles on PWR (Pressurized Water Reactor)—a 60 megawatt reactor for central station electric power. Reactors of this size, and perhaps larger, would be required for the propulsion of fast passenger ships.

Once a reactor becomes critical and starts operation, its level of power is not necessarily limited by nuclear characteristics. Its power is proportional to the fission rate, which in turn is proportional to the flux of neutrons scrambling to initiate fission. The neutron flux varies with the position of the reactor controls. The only operating limitation is the rate at which the fission heat can be removed from the reactor core. The faster the coolant can be pumped through the reactor, the more steam is generated outside. More steam available to the turbine, of course, means more power to the shaft, more speed, and so on. No matter how much steam is generated, the physical weight of fuel inventory remains unchanged!

The length of time that a ship can steam on its original fuel inventory will depend entirely on nuclear burnup—or nuclei depletion—characteristics. This might be in terms of six months or one full year. For example, the "X-10 Pile" at Oak Ridge, Tennessee, was started in November, 1943. It has operated greater than 90% of the time on its original fuel supply. Its nuclear depletion rate is less than 1% per year. The "X-10," of course, is a very low powered research reactor wholly unsuitable for ship propulsion purposes. Nevertheless, it illustrates the endurance potentials of nuclear power.

HIGHER CARGO-TO-FUEL RATIOS FOR SUPERTANKERS

In the article, "Rivalry on the High Seas" (Oil Forum, March, 1954), the author selects three supertankers, each of 30,000 deadweight tons, as being... "The most economical steam tankers in commercial service today." One of these ships on her maiden voyage developed over 16,000 shp. The fuel consumed... "was at the rate of 85 tons per day... at a speed of 18.6 knots." For a 6,000 mile haul (one way), this represents approximately 2,800 tons of fuel for the round trip. At 8,500 miles, the fuel requirements would be 3,800 tons. Thus, the cargo-to-fuel transported is 10.6:1 and 7.9:1,

respectively. Older ocean-going tankers have corresponding lower ratios, generally around 5:1.

The latest supertankers develop less than the shp of either one of the nuclear powered submarines. However, for illustration purposes, if we assume that the combined weight of the nuclear fuel plant (fuel plus reactor plus coolant plus shielding) of one of the subs is one-fourth its total displacement, we arrive at a fuel system weight figure of about 800 tons. This is not authoritative, but it is believed to be a conservative estimate for 3,500-ton submarines. If the submarine nuclear plant could be directly substituted in one of the "most economical" tankers, the resultant cargo-to-fuel ratio would be 37.5:1 (30,000 tons cargo÷800 tons fuel system). Compared to the best tanker ratio of 10:1, nuclear power would result in almost a fourfold increase—to say nothing of the additional cargo that could be carried for the diminished fuel tonnage.

Large passenger ships require even greater amounts of fuel than supertankers. The "America" and "Independence," for example, have fuel oil capacities of 5,000 and 8,000 tons, respectively. The "United States"—the fastest and one of the largest merchant ships in the world today—consumes better than 50 tons of fuel oil *per hour!* She has a displacement of approximately 45,000 tons and is propelled by approximately 175,000 shp to attain a speed of "approximately" 35 knots. The proportionate decrease in nuclear fuel system weight for a ship of this size is really an attractive consideration. But nuclear fuel costs are prohibitive, by Bunker C standards.

A 5:1 COST ADVANTAGE WITH BUNKER C

Despite the impressive potentialities of nuclear power, ordinary Bunker C fuel oil—used commonly on merchant ships—still has a gapping cost advantage over nuclear fuel. It will take a few paragraphs to point this out.

The real cost of nuclear fuel is unknown. Nevertheless, in "open literature" one can find quotations of cost at . . . "$15 to $20 per gram," or . . . "at least $20,000 per kilogram." Rounding these figures and converting into pounds, the approximate cost of nuclear fuel would appear to be $9,000 per pound.

Bunker C fuel oil costs around $2 per barrel. Approximately 7,500 bbls. of this fuel gives the equivalent heat energy of one pound of nuclear fuel, when completely fissioned. Thus, we would be getting about 40 billion Btu's of heat from $9,000 worth of nuclear fuel, as compared to $15,000 worth of conventional marine fuel. On the surface, this looks like an immediate cost advantage for nuclear fuel—but wait!

The $9,000 nuclear fuel is *pure* fissionable material; that is, every nuclei in it is theoretically fissionable. There are septillion nuclei (one followed by 24 zeros) in one pound of such material . . . occupying slightly more than one cubic inch. Each of these tiny nuclei, when fissioned, develops its own share of kinetic energy. It is a physical impossibility to effectively convert each and every fissionable nuclei into useful heat from a mass so small. Imagine, the heat equivalent of 1,000 tons of fuel oil (7,500 bbls.÷6.76 bbls./ton) from one cubic inch! It couldn't be controlled. The consequence is that pure fissionable material would *never* be used as fuel in a reactor.

The only successful fission heat-

transfer process involves less-than-pure nuclear fuel, either in the natural or enriched state. These less-than-pure fuels are much larger physical masses, with many less fissionable nuclei. For example, one pound of natural uranium contains only 0.7% of fissionable nuclei. Hence, 140 pounds are needed to give the same heat content as one pound of pure fuel. Published figures say that natural uranium costs about $35 per pound. This works out to about $5,000 which is even more attractive than pure fuel. Wait again! The catch is that only about 1% of the natural fuel can be burned up—an inherent nuclear limitation imposed by "self-poisoning" characteristics. So, actually, we need 100 times the 140 lbs. to produce our 40 billion Btu's of heat. In dollars, this is $500,000—a conspicuously unattractive figure. The use of "enriched" fuels greatly reduces this cost value.

Enriched fuel contains a greater-than-natural percentage of fissionable nuclei; say, 1%, 2%, 5% or higher. The higher the enrichment, the more fuel production (gaseous diffusion) difficulties encountered, such that 5% is about the practical limit for reactor fuels. Enriched fuel possesses the very definite advantage of higher burnup, by overriding the self-poisoning phenomenon. Open literature "guesses" that the burnup might go as high as 20%. If we extrapolate between our $9,000 pure fuel and our $5,000 natural fuel, 5% enrichment might cost $5,500. If we assume that less than 20% can be burned up, our equivalent fuel cost amounts to about $30,000. Compared to Bunker C at $15,000 enriched fuel *appears* to approach the realm of economic feasibility.

One more factor enters the fuel cost picture; namely, the comparative thermal efficiencies of reactors and boilers. Modern merchant vessel boilers, in operation, have thermal efficiencies of 85% and above. The best nuclear reactor efficiencies published for proposed reactors (not yet built) are 35%. Thus, to have an equal amount of heat for the generation of steam as in a boiler, a nuclear reactor requires two and a half times its initial fuel supply. This jumps the cost advantage of Bunker C from 2 to 1 in the preceding paragraph, to five-to-one. This five-to-one cost advantage of conventional fuel would completely discourage any consideration of nuclear powered merchant ships, were it not for government ownership and salvage of reactor fuels.

AEC CAN "LEND-LEASE" NUCLEAR FUELS

The Atomic Energy Act of 1954 reaffirms that the Government has exclusive ownership to all nuclear fuels. Section 52 of the Act says that . . . "All Rights, title and interest in or to any special nuclear material within or under the jurisdiction of the United States, now or hereafter produced, shall be the property of the United States and shall be administered by the (Atomic Energy) Commisison." The Act goes on to say that the Commission is authorized . . . "To issue licenses . . . to make available for the period of the license, and to distribute special nuclear material within the United States to qualified applicants. . . . The Commission may make a reasonable charge . . . (based upon) (1) the extent to which the use will advance . . . the development of peaceful uses of atomic energy; and (2), the energy value . . . (used)." From these statutory excerpts, it is clear that nuclear fuel for merchant ships would

have to be "leased." It could not be bought at any price.

The greatest percentage of fuel in a reactor remains unburned. Remember, a certain proportionately large amount is required for criticality . . . to make the reactor "go." When the excess fissionable nuclei are depleted below the amount for criticality, the reactor has to be completely recharged. The high first cost of nuclear fuel to the Government and its importance to national conservation, make it mandatory that the unburned fissionable nuclei be salvaged and reprocessed. However, if the fissionable nuclei depletion is below that in the natural state; i.e., less than 0.7%, salvage processing is much too uneconomical and the "waste" material is disposed.

The "fair price" lease-charge by the Government for the energy value of the nuclear fuel consumed would take many factors into consideration. For example, the actual Government cost of production, handling and reprocessing would be involved. However, it is conceivable that merchant ship operators could reduce this "fair price" by emphasizing reactor designs which bred *new* fissionable material. One of the outstanding features of nuclear fuel is that new fissionable nuclei can be bred from certain "fertile" materials . . . without detriment to the propulsion power load. Presumably, the new fuel would be bought back by the Government, as the Act prohibits private ownership of substantial quantities of such fuel and mentions (Section 56) . . . "guaranteed fair prices for all special nuclear material *delivered to* the Commission . . ."

FUEL RATE IN PRESENT BOILER TECHNOLOGY IS LIMITED

The extent to which nuclear fuels would advance the development of peaceful uses of atomic energy is one of the underlying premises of the 1954 AE Act. If it could be shown that nuclear power would push aside present technical limitations on merchant ships, and accomplish things that fuel oil couldn't possibly do, regardless of cost, then a good case would exist for merchant ship nuclear power.

The customary index of merchant ship propulsion plant efficiency is "fuel rate." This expresses the number of pounds of fuel consumed per shaft horsepower per hour (lb./shp/hr.). For modern vessels, the fuel rate ranges between 0.50 and 0.60 lb./shp/hr., the newest and most efficient plants being very close to 0.50. The continuing effort to improve this degree of operating efficiency has met with diminishing returns.

An extensive study of fuel rate and its variables is covered in the 1948 Transactions of the Society of Naval Architects and Marine Engineers. Refer to article, "Economic Selection of Steam Conditions for Merchant Ships." The authors have developed a group of curves showing per cent gain in fuel rate for various steam conditions, with corresponding curves showing the increased cost for accomplishing the fuel rate gain. To exemplify the effect of present boiler technology limitations, a ship of 12,500 shp at 600 psi (850° F) gains about 1% by raising its steam pressure to 850 psi (850° F). With this minor gain, increased complexity of the plant results.

The reference article concludes that: "Higher steam conditions require increased alertness and more attention to good housekeeping . . . (by) the operating crews. High pressure boilers are more sensitive to feed-water contamination, and internal and external

fouling of heating surfaces. Higher temperatures make it more difficult to maintain satisfactory joints of all types and to avoid valve stem and bushing problems. . . . Also, higher pressure systems require greater desuperheating and pressure reducing equipment for auxiliaries. . . . There is no general acceptance of the theoretical advantages to be gained by the more costly installations for higher steam conditions."

The trend in past quarter-century has been decidedly towards higher boiler steam conditions, for higher fuel efficiencies. There are some marine power plants operating at 1000 psi, 1000° F, achieving a fuel rate of 0.46 lb./shp/hr. This fuel rate is just about the practical—even theoretical—limit. The question now arises, "Where can attention be focused to seek still further gains in efficiency?" Perhaps, nuclear power is the answer. No one really knows.

IN SUMMARY AND CONCLUSION

Technical advances in search for more efficient operation of larger, faster merchant ships comprise the ever-receding goal in marine engineering. A typical supership of today gulps from five to fifty tons of fuel per *hour*, in a trend of rising fuel oil costs. Ingenious effort to reduce this extraordinary fuel consumption has met with definite practical limitations. Yet, still the competitive drive to haul more cargo and passengers, at faster speeds, over greater distances, persists. While the logical next step may be nuclear power, it will not come on its own accord. Nor will successful developments by the military and the electric utilities necessarily resolve all of the technical and operational problems involved. The time is approaching for selective thought, persistent effort and step-by-step evaluation by the merchant marine industry itself.

It has been said that if the American Merchant Marine is to become the dynamic world influence of which it is capable, "Our merchant ships must seek some achievement which significantly contributes to the betterment of Man." Conceivably, the application of nuclear power would be the greatest achievement ever.

AN ADVENTURE IN COOPERATION
Integration of Government and Industry Standards

R. V. VITTUCCI

THE AUTHOR

After receiving the ME and MME Degrees from Cornell University, the author began his professional work in Heat-Power at the American Gas and Electric Company, New York City. For several years he taught Mechanical Engineering at the University of Texas, where he published a laboratory manual. He has been with the Navy Department, Bureau of Ships, since 1951, except for three years with E. I. duPont de Nemours and Company. His early work in the Bureau was in the Heat Balance and Ship Trials Branch. A registered professional engineer in the state of Delaware and in the District of Columbia, he is presently serving in the Bureau's Standardization Planning Branch.

The following paper formed the basis of a speech which he presented at the Third National Convention of the Standards Engineers Society in Atlantic City, 1 October 1954 before representatives of government and industry.

YOUR MONEY'S INVOLVED

Every working day of the year the U. S. Department of Defense spends on the order of 77 million dollars for material and equipment. At the same time, this Department invests about 52 thousand dollars a day (over $100 every minute of the work day) in standardization programs which lead to the specifications and standards needed for this procurement. The money is furnished by you and me, the citizens. It's a big investment but it is paying big dividends.

More and more does our government realize the tremendous value of realistic standardization—not the kind done in an ivory tower, but rather the dynamic kind with industry considered as a partner. Our legislators know only too well that something has to be done to reduce the staggering burden on taxpayers. Needless waste, caused by duplication of effort, the high cost of so-called "specials," confusion due to fantastic variety of items in government stocks which contain both the specials and the standard commercial products—all these can not be tolerated. This is especially true in view of the mammoth requirements for equipment in the event of war.

STANDARDS, STANDARDS!

All about us we find standards—industry, or commercial standards such as those of ASA, ASTM, NEMA, RETMA, ASME, AIREE, etc., company

standards, and government standards. It is no wonder that some of these overlap or conflict.

The U. S. Department of Commerce, which includes the Bureau of Standards, has as one of its functions the duty to assist, coordinate and cooperate in the creation of voluntary industry standards. This may appear to conflict with the stated purpose of many commercial organizations already dedicated to that cause. However, by a recent directive, that Department issued a policy which will permit carrying out this function without competing with or duplicating the efforts of industrial standardization bodies which could handle the work.

WHAT IS INTEGRATION?

With many engineers in government and in industry working to find better and cheaper ways of doing things, a certain amount of "integration" of standards of these groups will inevitably take place. By integration I mean the adoption of one another's standards, wholly or in part. Stated another way, I mean including industry standards in government standards or specifications—and vice versa.

This differs from unification, which suggests only one series of documents to be used by both industry and government. Unification of this sort is actually done in some countries but, under the United States system of Government, it is not likely to occur here. Integration, however, has already taken place to a certain extent and there is a strong trend towards more of it to the ultimate benefit of all concerned. The word "standard" is used in its broadest sense here and includes specifications and any other standard-type document.

TO EACH HIS OWN

With all the thousands of perfectly good industry standards available, why do government agencies find it necessary to write their own? There are many good reasons for this. Here are a few:

1. For some items there aren't any industry standards. This is particularly true for military equipment such as guided missiles.

2. Industry standards often define products which are inadequate for military use. Equipment built to commercial standards is seldom subjected to the intense shock and other rough treatment given to naval and other military epuipment.

3. Industry standards are not controlled by the government and, therefore, may be changed at any time without government concurrence. Control over revisions is necessary in order to preserve reasonable uniformity of govenrment equipment.

4. Difficulty is frequently encountered in obtaining copies of such documents to furnish to a large number of bidders and inspectors.

5. Some non-governmental specifications have contract requirements which may conflict with the legal procedures of the Federal government.

6. Government purchasing policies make it mandatory to have open competitive bidding. Federal agencies must accept the lowest satisfactory bid even though it is sometimes known in advance that a slightly higher price would bring a much better bargain. For protection the Government must have clear, tight specifications and standards.

GOVERNMENT POLICY ON INTEGRATION

In 1949 the General Services Administration (GSA) was created and, within its Federal Supply Service, there was established a Standardization Branch. That Branch is:

> responsible for the development and maintenance of standard purchase specifications and a system of inspection and testing of executive agency purchases, including direct inspection of GSA purchases, standardization of commodities purchased and acceptable products lists.

In 1951, at the direction of the President of the United States, the General Services Administration and the Department of Defense made what they called an "Area of Understanding Agreement." They included in the agreement these significant statements of basic policy:

> 1. A Federal or a Military specification or standard will not be prepared when it is not in the best interest of the Government to do so.
>
> 2. Nationally recognized industry and technical society standards and specifications shall be used to the maximum extent practicable in the development of Federal and Military Specifications and Standards. Normally, this will be done by adopting by reference or by transcription from such industry and technical society standards or portions thereof, and issuing the results as coordinated or Interim Federal or Military Specifications or Standards, without deviation unless the exception can be justified. When the requirements are essentially similar to those contained in a nationally recognized standard, an appropriate note to this effect should be included in the specification or standard.
>
> 3. The General Services Administration and the Department of Defense shall establish and maintain relationships with approved industry groups as required in order to coordinate the development of specifications, standards, packaging requirements and inspection procedures with the best practices of industry and to obtain to the greatest extent practicable the cooperation and participation of industry in the program.

These policies, which are still in effect, put the Government clearly on record as favoring the use of recognized industrial specifications and standards whenever they will satisfy the Government needs. They not only save time within the Government but also assure that our civilian and military requirements are properly geared to current industrial practices and resources.

NARROWING THE PROBLEM AREA

Examination of the Federal expenditures shows that the cost of national security has constantly dominated the budget, requiring approximately two-thirds of the total national expenditures. Of this portion, the amount estimated for procurement of new aircraft, ships, tanks and weapons—including repair and maintenance parts—totaled about 20 billion dollars in fiscal year 1954. Even with the economies of fiscal year 1955, this amount will be 17 billion dollars. These enormous quantities when compared to the entire Federal budget clearly indicate where to strive for savings. Better standards and specifications for the Department of Defense procurement and maintenance are the most promising steps towards economy.

Eager to secure the benefits of standardization, the 82nd Congress in 1952 passed Public Law 436 known as

the Defense Cataloging and Standardization Act. One of the purposes of this Act was to unite and intensify the standardization efforts within the Department of Defense. This law set up the Defense Supply Management Agency (DSMA) to carry out this objective. Under Reorganization Plan No. 6, 1953, this Agency was abolished and its duties transferred to a new Assistant Secretary of Defense for Supply and Logistics. The Office of Standardization, as it is now called, has the important task of planning and coordinating the Deparment of Defense Standardization Program within the armed services. The actual standardization work is farmed out to the various military departments. Each department is assigned whole groups of related items called "classes" in the fields where it has the greatest experience.

A CLOSER LOOK—THE WORKING LEVEL

The technical standardization know-how in the Department of Defense is centered in the various divisions of the Army, Navy and Air Force. These departments get the work done. As an example, let's look at the Bureau of Ships of the Navy Department.

Bureau representatives participate in a complicated network of standardization activities, including the following: the military program of the Department of Defense, the industrial programs of national groups like the ASA, ASTM, SAE, and many other technical societies. Added to this are international military programs with the North Atlantic Treaty Organization (NATO) with the British and Canadians (ABC Standardization), and industrial standardization with the International Organization for Standardization (ISO) and the International Electrotechnical Commission (IEC).

Fully realizing the importance of accepting industrial standards, the Bureau makes every effort to take advantage of help offered by industry. All new standards or specifications are coordinated not only with other interested Bureaus and departments of the Government, but also with segments of the industry concerned, including producers and suppliers. Each industry is thereby alerted at an early stage to new requirements of the armed services. Industry representatives point out the possible increase in costs, procurement delays, need for retooling or new processes, as well as the effect upon free competition. In this way the ability of industry to produce the quantity and quality required is determined before the final decision on a standard is made.

Both large and small companies, including trade association members and independent companies, are contacted. Coordination is brought about by calling upon industry coordination committees or by circulating proposed drafts for comment. In fact, drafts are sometimes submitted directly to trade associations for comment. Every effort is made to reconcile the Government requirements with the recommended commercial standards. The Armed Services Electro-Standards Agency (ASESA), which coordinates specifications and standards of electronic components for the three services, is in constant touch with industry.

HELPING SHAPE INDUSTRY STANDARDS

Probably the greatest number of direct contacts with industry standards comes about through Bureau of Ships or Navy representation on commercial standardizing bodies such as the ASTM, ASA, ASME, SAE, etc. This

Bureau has always cooperated liberally with industry and technical societies on these matters. In fact, the Bureau has representatives on over 500 committees and sub-committees under the sponsorship of only a dozen of these organizations. It is also indirectly represented in other groups by virtue of the personal memberships held by certain employees.

Until recently active government participation in such organizations was a moot point, particularly when voting was involved. In September 1954 a new directive cleared up this uncertainty and defined the extent of participation permitted to government agencies. In brief, government agencies may participate and vote on technical matters but not on industry politics.

SOME ACCOMPLISHMENTS UNDER DEPARTMENT OF DEFENSE STANDARDIZATION PROGRAM

An illustration of the kind of industry-government cooperation that has been going on in this program is the work being done by the Department of Defense Ad Hoc Committee on Standardization of Steel Specifications. Some time ago the Navy recognized disparities in government and industry specifications for steel. The Office of Standardization organized a committee consisting of representatives from the Army, Navy and Air Force, the General Services Administration, the American Iron and Steel Institute and the Office of Standardization to bring about some standardization in the many types and grades of steel products used by the Government. The overall plan of this committee was to establish three up-to-date documents which, in turn, could be referenced into any iron or steel specification. These three basic documents are:

1. "Tolerances for Steel and Iron Wrought Products" Federal Standard No. 48.
2. "Steel; Chemical Composition and Hardenability" Federal Standard No. 66.
3. "Test Methods for Metals" Federal Specification QQ-M-151.

The first two of these documents have been issued. The third is still being worked on and will be issued as Federal Standard No. 151. These standards, together with the handbook of "Steel and Iron Wrought Products" provide the foundation for revision of approximately 250 steel specifications. They will help bring them all into line with the current standard industrial practices.

Other large programs include standardization of maintenance parts for internal combustion engines and standardization of abrasive wheels as well as some proposed studies in the fields of plastics and rubber products.

Every six months, according to law, reports of savings are made by all the military services to the Office of Standardization where they are summarized into a single Department of Defense report to Congress. Among the savings reported are these:

Adoption of commercial packaging for sensitized photo material resulted in annual saving of $50,000.00 to the Navy Bureau of Aeronautics.

The Army Quartermaster Corps adopted commercial box lockers in lieu of specially designed military lockers. Savings during 1951, 2 and 3 were $7,605,000.00.

The total of the individual savings for the whole Department of Defense is shown by the following summary of benefits for the first six months of calendar year 1954:

Eliminated 45,000 Unnecessary Items from Supply System.

Consolidated 174 Separate Specifications into 59.

Saved 120,000 Cubic Feet of Storage Space.

Saved 400,000 Man-Hours.

Saved $65,000,000.

WHAT ABOUT INTEGRATION?

What progress has been made towards integration of government and industry standards? Complete information on this subject is practically unobtainable at present. However, a study made in the Bureau of Ships during August 1954 reveals a list of 266 Military or Federal Specifications and Standards which include and specifically reference 222 different commercial standards. (See Tables 1 and 2.)

A survey of this subject in the Army, Navy and Air Force reveals a list of at least 60 industrial specifications and standards that are used by two or more of these departments in their documents. These figures hardly begin to indicate the true extent of integration that has already taken place between government and industry standards because they include only those cases where the industry standard was actually named in the Government document.

Consider Federal Standard VV-L-791 which shows methods of inspection, sampling and testing of lubricants, liquid fuels and related products. This book starts by listing each test method, telling whether it is identical, similar to, or different from the corresponding commercial standard. About 50% of the methods used are identical to ASTM standards. This document is an excellent example of the type of integration about which we are concerned.

TURN ABOUT IS FAIR PLAY

And what about industry? Does industry ever make use of government specifications and standards? Considering the magnitude of government purchases, especially in time of war, men in industry find it good business to watch closely any moves made by Uncle Sam to set new standards. A manufacturer in the early stages of his standardization program may find it profitable to obtain any existing government specifications or standards pertaining to his field. Much engineering time and experience has gone into producing these documents which may be obtained at a very small cost from the Superintendent of Documents, Washington 25, D. C.

Consider Military Standard 210 "Environmental Extremes" which is used by all three services and has recently been adopted by the military establishments of many NATO countries. This document has a wealth of information concerning climatic conditions existing all over the world, such as temperature, rainfall, relative humid-

ity, etc. It tells, for instance. how dry it is in the Sahara when it is very, very dry and how cold it gets at the North Pole. The only thing it does not tell is how high is "up." This factual information serves as a valuable guide to designers who must know the worst climatic conditions their equipment might have to encounter. The cost of measuring and assembling this information with the accuracy found in this document would be prohibitive for a private company, yet the information is available to the public for little more than the asking.

The Radio and Electronic Tube Manufacturing Association (RETMA) is using Manuals M5 and M6, the military outlines of form for preparing specifications and standards, as formats for writing their own industrial specifications on electronic items.

Other documents of a general nature are available, such as the Federal Supply Classification, which arranges all items in the Federal supply into related groups. This document together with several others relating to identification of items could serve as a basis for establishing a catalog or commodity classification system in an industry about to make one for themselves.

THE NECESSARY ELEMENT—COOPERATION

The 1955 military procurement costs will total more than 17 billion dollars. It is clear that even a small fraction of 1% saved through good standards would be worthwhile. One of the ways to save this money is by working for the optimum degree of integration of government and industry standards. The continuing trend of government and industry towards the voluntary use of common standards will not only save money for everyone, but it will also help keep this nation and her industries strong enough to preserve our superior standard of living.

We have the *incentive* for going on. The administrative *machinery* for accomplishing this end already exists in the elaborate array of government and commercial standardizing bodies. We have the engineering *talent*—our country has standards engineers as fine as any in the world. We have the *money* to carry on programs like these. The final element necessary to success in this dynamic adventure is *cooperation*, the lubricant which keeps that machine in smooth operation. Judging from the remarkable industrial progress in this nation, we certainly have enough of that!

TABLE I

COMMERCIAL STANDARDS USED AND REFERENCED IN 266 GOVERNMENT SPECIFICATIONS

87	standards of	American Standards Association (ASA)*
60	" "	American Society for Testing Materials (ASTM)*
23	" "	Society of Automotive Engineers (SAE)*
17	" "	Underwriters Laboratories (UL)
5	" "	American Society of Mechanical Engineers (ASME)*
1	standard "	American Welding Society (AWS)*
3	standards "	American Institute of Electrical Engineers (AIEE)**

1	standard	of American Chemical Society (ACS)**
1	"	" American Society of Refrigeration Engineers (ASRE)
4	standards	" American Iron and Steel Institute (AISI)*
1	standard	" Academy of Motion Picture Arts and Sciences (AMPAS)
1	"	" National Board of Fire Underwriters (NBFU)
2	standards	" National Electrical Manufacturers Association (NEMA)*
3	"	" Association of American Railroads (AAR)
1	standard	" American Railway Engineering Association (AREA)
1	"	" Lithographic Technical Foundation (LTF)*
2	standards	" American Gear Manufacturers Association (AGMA)
1	standard	" Textile Color Card Association (TCCA)
1	"	" Lowell Textile Institute (LTI)
4	standards	" American Wood Preservers Association (AWPA)**
1	standard	" National Hardwood Lumber Association (NHLA)
1	"	" Southern Pine Inspection Bureau (SPIB)
1	"	" West Coast Lumberman's Association (WCLA)
222	standards	TOTAL

*Bureau of Ships has membership.
**Individual memberships; employees represent Bureau on committees.

TABLE II

NUMBER OF GOVERNMENT SPECIFICATIONS
IN WHICH COMMERCIAL STANDARDS ARE USED

95 use ASA Standards	4 use NEMA Standards
58 use ASTM Standards	2 use NHLA Standards
6 use AISI Standards	1 use SPIB Standards
5 use AIEE Standards	1 use WCLA Standards
1 use ACS Standards	3 use AWPA Standards
1 use AMPAS Standards	4 use TCCA Standards
23 use UL Standards	1 use LTF Standards
5 use AAR Standards	1 use LTI Standards
1 use AREA Standards	1 use AWS Standards
6 use AGMA Standards	1 use ASRE Standards
11 use ASME Standards	2 use NBFU Standards
33 use SAE Standards	

Total number of Military and Federal Specifications in the above group is *266*.

DEVELOPMENTS IN AIRCRAFT ELECTRICAL POWER

ACKNOWLEDGEMENT

This article, by G. A. Phillips of General Electric Company, was published in the October 1954 issue of *Aero Digest*.

The airplane of today is dependent upon electric power. No longer can the electric system be considered as a minor system used to charge batteries for lighting and radio. Today it is all-important as the nerve system and the sinews to perform functions, which with the advent of near-sonic and super-sonic planes, can no longer be performed manually by the pilot or the crew. In addition, the functions of navigation, communication, offense, and defense will usually not be possible without the supply of adequate electric power. Many new aircraft when flying under instrument conditions either cannot maintain control when the generating system is lost, or have less than one-half hour operation on the battery before control is lost. In other words, the modern airplane may not even be able to return and land safely without electric power.

This dependence on electricity has come about primarily because of the increasing number of functions for which this type of power is either essential or best adapted. Among the hundreds of items of electrical equipment found on a large airplane there may be in excess of 100 electric motors as on the B-36. The extended use of electricity on aircraft got its first big impetus during World War II. From a few kilowatts in 1940, the aircraft electric system had grown by 1950 to over 100 kw installed capacity on medium bombers. There has been a tendency since 1950 for the rapid growth to level off. This is due, no doubt, to recent trends in the industry toward simplification and limitation in aircraft physical size. It is true that the installed capacity on very heavy bomber, such as the B-36 and the B-52, is in the neighborhood of twice the figure for a medium bomber. Again, however, it would appear that for aircraft of this type there is some tendency towards saturation in the use of electric power.

Medium bombers, as represented by the Douglas B-66, have in the neighborhood of 90 to 120 kva. The Douglas DC-7, however, is somewhat lower, this being typical of the present-day large commercial transport. There has been no major electric power increase in this category since 1948 with the advent of the Boeing *Stratocruiser*. Our studies show that the first U. S. jet transport, however, will undoubt-

edly have 120 kva or greater capacity, placing it up with the medium bombers again.

Possibly of greatest interest are the figures for fighter aircraft. One of the new long range escort fighters approaches the commercial transport in utilization of electric power for the many functions which must be performed. Even the smaller fighters, such as the McDonnell F3H and the Lockheed F-94C, use large amounts of electricity. It is interesting to note that in the case of the F-94C, 30 kva is required for de-icing, a function not generally performed electrically at the present time. Further development in techniques for electric de-icing and in the means of applying heating elements to the surfaces may well lead to another large increase in electric power demand on other types of aircraft.

ELECTRIC LOADS

Electric power is used on aircraft to increase performance, economy, safety, comfort, and military effectiveness. The number of individual functions performed by electricity to accomplish these objectives has steadily increased over the years and has now become too extensive to mention in detail. However, the scope can be visualized by the following categories, each of which contains a large number of indivdual items:

Lighting, instruments, actuators, control systems, pumps and blowers, heating, communication and navigation, offensive and defensive armament, and flight control systems.

In addition to the total electrical demand represented by all these loads, it will be necessary to examine them for the type of power to which they are best suited in order to understand the present trends in electric power systems. With the possible exception of a few of the actuators and some control and warning systems, constant frequency ac power is the logical choice. While it is true that many individual items of equipment currently in production are designed to operate on the old 28-v dc system, most of the newer designs require 400 cycles ac. Also, communication equipment requires 400 cycle ac power and radar uses relatively large amounts of ac, which may be either variable or constant frequency. One of the largest areas of load growth has been in this field of electronics.

Motors, of course, have become one of the largest loads on those aircraft where nearly all functions are performed electrically. However, practices still vary widely among the airframe manufacturers in this regard, many using hydraulics for actuation of flaps, gear, and similar services. Therefore, the amount of motor load on various aircraft in the same category shows much variation. While it is true that dc motors of the same weight are capable of higher starting torques than ac motors, the latter by proper design and application may be used for virtually all aircraft needs except where closely controlled variable speed is required. However, many of these variable speed applications, such as turret drives, have a variable voltage control which employs a motor-amplidyne or other means of power conversion which is actually better suited to an ac power supply than to dc. AC induction motor drives are preferred wherever possible because of their simple and rugged construction with no brushes and lower maintenance.

TREND TO AC

The increasing use of alternating current is a dominant factor in the industry today. One of the principal reasons for this may be found in a study of the load equipment going on these airplanes. The preceding discussion of electric loads shows how well adapted alternating current is to the needs of aircraft systems. There is no doubt that this is the most significant element in the present trend to ac power generation. However, it is interesting to see how this trend has developed, why aircraft are configured as they are today as to power generation, and where the trend is leading.

Prior to World War II electric power was commonly supplied at 12-v dc with a battery forming a substantial part of the system capacity. With the increase in power requirements around 1940 is was clear that prohibitive weight penalties would result from the distribution of power at this low voltage. Experimental work had been sponsored by the military services on several systems including ac prior to 1940. However, with the demand for a rapid build-up in production required with the beginning of World War II, the best immediate solution was the widespread use of the 28-v dc system. Many loads have been developed for this type of power and it was relatively easy to convert others. Also, many of the older 12-v. dc aircraft could be easily modified to take the new 28-v. equipment. The writer had the experience during the war in the Southwest Pacific of converting a number of 12-v. C-47 aircraft to 28-v. This modification was developed in the field towards the end of the war primarily because of the increasing difficulty in obtaining spare parts for the 12-v. systems and equipment.

Through 1950, with few exceptions, 28-v. dc provided the bulk of aircraft electric power. This was supplemented in many cases by direct engine-driven, variable-frequency ac generators rated up ot 30 kva. These machines provided power of a type which could be directly used by heating equipment such as propeller de-icing, or by electronic equipment such as radar. Furthermore, since the 28-v. dc generators were limited to 12 kw output, it was possible to increase the generating capacity on the limited number of available pads provided on most engines. However, one of the disadvantages with this arrangement was the fact that increasing amounts of constant frequency 400 cps power was also required. This power has always been supplied in small amounts from the 28-v. dc system through inverters which are nothing more than dc to ac motor generator sets. By 1950 a number of aircraft required two or more inverters rated at 2.5 kva each. This represented a large weight penalty as well as a demand for more dc capacity, since the inverters are approximately 50 percent efficient. In addition, it is readily apparent that the airplane had become quite complex electrically, since there were now at least three separate types of power distribution i.e., 28-v. dc, variable frequency ac, and 400 cycles constant frequency ac.

Because of the increasing weight penalties associated with distribution at 28-v., 120-v. dc systems were considered with favor by many people in the industry during the period 1947-1949, and the development of generating equipment was sponsored by the Air Force. However, this system was not accepted for a number of reasons, most important being the preponderance of ac loads and the continued need for inefficient inverters to supply the needed constant frequency 400 cycle power.

Following closely on the 120-v. dc developments just prior to 1950 were other factors pushing the industry toward ac. Reasonable success with the secondary variable frequency system was significant but the foremost factor was the availability of several types of constant speed drives. The Sundstrand variable-ratio hydraulic transmission had achieved a satisfactory degree of serviceability on the B-36. Turboprop engines operating over a narrow range of speed provided means of getting substantially constant frequency from a direct engine drive. Turbine drives were made possible by use of the compressed air available from bleeding a jet engine compressor. These favorable factors and the Korean War led in 1951 to an accelerated development program undertaken by General Electric for the Navy of improved ac generators and voltage regulators. These were built to a new experimental specification prepared by the Bureau of Aeronautics which was later to become MIL-G-6099, the present ac generator standard for all military aircraft. Control and protective system developments for both single generator and parallel system were also undertaken by the manufacturers. The availability of these new drives, generators, and control, coupled with the increasing ac load demand, has led in the last two years to the application of ac power generation to all present day aircraft.

One of the most popular current combinations for small and medium aircraft is what we might call the two-system airplane, employing both 28-v. dc and constant frequency ac power generation. By the addition of the constant frequency generators it has been possible even with the additional drive weight to achieve an actual weight saving by replacement of the inverters and variable frequency generators along with a reduction in the dc capacity resulting from elimination of the inverter load. By providing an adequate supply of constant frequency ac it is possible to use induction motors with a further weight reduction and increase in reliability for many loads. As the dc load is reduced, it becomes economical in some cases to eliminate the dc generators entirely and supply 28-v. dc power by means of transformer-rectifier units operated from the ac system. These small and medium sized aircraft thus are beginning to approach the all ac system as originally developed during World War II for the B-36 and recently applied to the other very heavy bomber, the B-52. I fully expect to see the first commercial jet transport built in this country provided with all ac electric power generation.

GENERATORS

Although the application of ac power is increasing very rapidly, there are still a considerable number of aircraft going into production which require dc generators. Due to the higher power requirements on both military and commercial transport aircraft, it has been necessary for the dc generator designers to keep pace by improving the output within the same weight and space envelope. As an example, the General Electric Model 2CM244A2 machine has been put in service on the Douglas DC-7.

This generator incorporates a number of new design features which increase the output, improve the reliability, and reduce the operating cost. Due primarily to the addition of the Freon cooling system on the DC-7, its load was increased appreciably over that of the DC-6B. This resulted in the need for a generator with good base

speed characteristics capable of delivering 250 amp at 3100 rpm with one in. of water pressure drop for one-half hour during ground and taxi operation. It also required a generator capable of delivering 400 amp for intermittent periods up to 10 hours during take-off, climb, and cruise. The heating in the commutating and compensating winding was reduced by the use of a single circuit of rectangular formed bar copper rather than the four-parallel circuits of round insulated wire used previously. By use of the shrink ring commutator the ability of the commutator to withstand higher temperatures and higher speeds without deformation has been increased. A newly developed General Electric brush suitable for both sea level and high altitude operation was also used as it permitted high loads without increase in temperature. Furthermore, this new brush has the desirable characteristic of being able to operate at altitude without a prolonged run-in at sea level to produce the commutator film necessary with older types of altitude treated brushes. Thus it was possible to produce a generator with very little increase in weight and overhung moment over the earlier 300 amp rated machines yet capable of meeting the more severe requirements of the DC-7. Among the several improved mechanical features found on the new dc generator are the quick-attach-detach mechanism which eliminates the need for removing the nuts on the mounting studs which were very inaccessible in this installation.

All generator manufacturers have concentrated a great deal of effort on the design of ac generators for the many new applications which are steadily increasing in number. Typical of these machines is the General Electric machine built in an 11-in. frame dia., characteristic of a complete line of ratings from 15 kva to 90 kva. Most recent developments in this line are 20, 30, 40, and 60 kva, 5700/6300 rpm, 400 cycle, 208/120 v, .75 power factor, suitable for use from sea level to 65,000 ft. altitude. At sea level the machines are capable of operating at full load with 80 deg. C. inlet air temperature and six inches of water pressure drop.

Since some loads require good wave form, harmonic suppression has been accomplished by optimum pitch and distribution as well as skew in the stator. The third harmonic is non-existent for balanced loads from no-load to 200 percent load. No harmonic exceeds one percent over the rated speed range.

Transient response is good with a recovery time to normal voltage after applying or removing full load less than 0.15 sec. Since machines must operate in parallel and may also be subjected to severe transients due to synchronizing and short circuits, they have been designed with extremely low resistance amortisseur circuits. This means that when heavy loads are applied or removed, when system faults occur, or when synchronizing is accomplished, mechanical oscillation of the rotor is reduced and electrical transients are damped quickly. This also increases the stability of the governing systems of the constant speed drives when machines are operated in parallel.

The rotor construction on these ac machines requires careful consideration of a number of conflicting factors. In order to achieve maximum stiffness, the rotor punchings are stacked on a hollow steel outer shaft without the use of spiders to support the laminations. This also reduces possible fire hazard since electrical failure in the machine or a bearing failure will not result in mechanical disintegration of the rotor due to collapse of light weight spiders. The weight is concentrated near the mounting flange end in order to reduce the overhung moment

and increase the generator natural frequency. A dc exciter is mounted on the same shaft in order to provide the field excitation for the ac machine. The slip rings are located adjacent to the commutator on the outboard end so that all brushes are accessible under one brush inspection band.

The blast cooling air is introduced through the opening in the blast cap and passes through the ac field after passing over the exciter. On some models a fan is provided which induces enough air flow during ground operation when no blast air is available to enable the machine to deliver full output. A serious cooling problem has existed on earlier ac machines due to the tendency of the rotor to run hotter at altitude than at sea level. The altitude rating has been improved by reducing heating losses in the field and by careful design of air passages through the rotor which actually induce increased cooling air flow. As a result, it has been possible to obtain full output on a 60 kva machine both at sea level with 80 deg. C inlet air and six inches pressure drop and at 60,000 ft. altitude and zero deg. C with 10 in. of water pressure drop. Generators with this performance are generally suitable for airplanes having high subsonic or low-supersonic speeds.

GENERATOR CONTROLS

As in the case of dc generators, continual improvements are being made in dc generator control equipment. In the latest type of voltage regulator for 28-v dc generators, the regulator itself is a plug-in unit weighing less than three lb. which, together with the vibration mount, weighs less than five lb. It is required to dissipate 90 watts and operate over a resistance range of 1.25 ohms to 35 ohms while meeting the severe performance and environmental requirements of MIL-R-6809.

One of the most interesting features of the regulator is the center of gravity type vibration mount designed specifically for the carbon pile regulator, but also readily adaptable to other types of control equipment. It is unique in that it permits the protected device to be mounted in any plane and isolated from vibration in any direction. In addition, the design was required to transmit less than one g vibration when subjected to 0.060 in. total excursion from 10 to 60 cycles and 10g equivalent displacement from 60 to 200 cycles.

AC generators of aircraft design require several hundred watts of dc power for field excitation which is customarily supplied through slip rings on the rotor. While this power could be supplied from the separate dc system on airplanes which have one, it is considered better practice not to have the systems interdependent. Therefore, modern ac generator designs employ a small dc generator on the same shaft for this purpose. The ac output voltage is regulated by controlling the field current of this exciter.

The ac voltage regulator has employed the carbon pile principle for a number of years. But additional features which have been added are an ac sensing circuit, an accurate glow tube voltage reference, an electronic trim circuit, and a stabilizing transformer. Greatly improved voltage regulation over that of the basic carbon pile is obtained by the electronic trim feature operating against a reference which is much more accurate than the spring and solenoid of the carbon pile. The

design is such that a failure of the trim circuit does not cause failure of the regulator, but simply gives the somewhat poorer voltage regulation of the carbon pile.

Although the carbon pile regulator has given a good account of itself in both dc and ac aircraft systems, it has several disadvantages among which are: short life due to wear of the carbon pile, difficulty in providing good temperature compensation, and effect of linear acceleration on the stability and regulated voltage. Therefore, at the time the General Electric Company undertook to develop improved ac generators and regulators for the Bureau of Aeronautics, it was decided to go to a static magnetic amplifier type of regulator which would eliminate these problems and give in excess of 5000 hr. life without maintenance. The ac output voltage is compared with an accurate voltage reference and the error signal fed into a two-stage magnetic amplifier which supplies the exciter field current. The original regulator developed for the Navy used a bias field in addition to the main exciter boost field since it was necessary to operate with variable frequency generators. The bias field provides stable operation over a wide range of main field current.

The latest development in static regulators has been developed for the Air Force and will be used on the Douglas B-66 as well as on other advanced types of Air Force planes. The principle of operation is the same except that no bias field is required, since the regulator has to operate only with constant frequency generators.

This new regulator represents an important advancement since its long life eliminates the need for the plug-in feature used on the carbon pile to facilitate maintenance. Furthermore, the use of static components permits the regulator to be bolted directly to the airplane structure without the use of vibration mounts thereby saving considerable space and weight. These regulators must also make provision for parallel operation of ac generators by dividing reactive loads equally among machines.

In order to provide power to the airplane electric system with convenience, safety, and reliability, the power must be controlled by suitable switching and the generator and power system adequately protected. A great deal of progress in the control and protective functions to be applied to both single and multiple ac generator installations has been made in the past year. Switching has been made completely automatic so that in a twin-engine airplane all the pilot has to do is start the engines. Generators come on the line and parallel automatically, also disconnecting the source of ground power. Depending on the requirements of a particular application, protection may be provided for any of the following conditions, selectively removing and de-energizing the faulty generator if operating in parallel: Short circuits, over or under voltage, over or under excitation, open phases, under speed or under frequency, or reversed phase sequence.

These control and protective functions together with a source of dc control power, usually a small transformer-rectifier, are often grouped on a single control panel. This particular panel is designed for parallel system operation and is of drawout construction to be mounted on a rack in the airplane. Other similar units are designed to be bolted directly into the airplane and there are also installations where, due to limitations of space, it is desirable to mount various components separately.

As mentioned earlier, it has been found very desirable in a number of airplanes to supply 28-v dc power from the ac system rather than have a separate dc generating system. This is customarily accomplished by the use of a transformer-rectifier through which the voltage is reduced to the desired level and rectified by means of selenium rectifier stacks. These units have been built in both regulated and unregulated versions and in ratings from a few amperes up to 200 amp. Typical weights are 17.5 lb. for a 50 amp unregulated unit, and 70 lb. for a 200 amp regulated unit. There will undoubtedly continue to be a need for transformer-rectifiers to supply 28-v dc power as more and more aircraft are built with only ac power generation. It is to be expected, however, that the amount of low voltage dc needed will steadily decrease as more and more load equipment is designed for ac operation.

Both dc and ac generators are now being made available which can operate successfully with cooling air temperatures greatly in excess of the 80°C. mentioned earlier. This is being accomplished by the use of higher temperature insulating materials, improved electromagnetic design for minimum losses, and refinement in air passages and heat transfer techniques. Studies are also showing that it will be economical to use oil for generator cooling since oil is already used for cooling and lubrication in the engine and in both the hydraulic and turbine type constant speed drives. Generator manufacturers have already built and successfully tested, both dc and ac oil cooled machines. Such environment-free machines remove the limitation of the high temperature, low density cooling air from the generator installation although, of course, it is necessary for the airplane manufacturer to provide means of disposing of the heat transferred to the oil. However, this problem is already present and by combining the cooling functions for more than one piece of equipment, better overall performance is often possible.

Weight reductions are being accomplished by investigation of higher rotational speeds and by refinement of electromagnetic and mechanical design. For small aircraft in the fighter category, General Electric has developed a line of 7-in. frame diameter ac generators in a range of ratings from 5 kva to 20 kva. These machines have either an integral exciter or if needed, can be provided with an external static excitation system, in order to reduce weight and space in the type engine installations created by the demand for smaller frontal area.

In the area of control equipment, relays, capacitors, rectifiers, transformers, and reactors are available for ambient temperatures in excess of 120 deg. C. To minimize the environmental problem, particularly altitude, many types of relays and other components have been hermetically sealed. By miniaturization of components and improved manufacturing and assembly methods, both space and weight are being steadily reduced.

As the needs of aircraft for more and more electrical equipment with improved performance have faced the electrical equipment manufacturers, solutions to the design problems have been and are being found. The many types of load equipment now required in aircraft in order to accomplish its intended purpose are dictating a rapidly increasing use of 400 cycle 3-phase ac power. Equipment is currently available for supplying this type of power on all types of airplanes and its capability for operating on higher performance planes is being steadily improved.

RECENT DEVELOPMENTS IN BOILING RESEARCH

W. H. JENS and G. LEPPERT

THE AUTHORS

Wayne H. Jens was born in 1921 and graduated from the University of Wisconsin in 1943. He was employed by North American Aviation until 1944, when he was commissioned in the Naval Reserve and served on active duty until 1946. He received the M.S. (1948) and Ph.D. (1949) degrees in Mechanical Engineering from Purdue University. As a member of the Reactor Engineering Division of Argonne National Laboratory from 1949 to 1952, he was engaged in heat power research and development leading to the design of the nuclear power plant for the USS *Nautilus* and was a group leader in the division. Since 1952 he has been with Nuclear Development Associates, White Plains, N.Y., as a Senior Mechanical Engineer.

George Leppert was born in 1924 and graduated from the University of Wisconsin in 1947. He served in the United States Navy from 1943 to 1946 and is a qualified submarine officer. He received the M.S. (1952) and Ph.D. (1954) degrees in Mechanical Engineering from Illinois Institute of Technology and was engaged in the Reactor Engineering Division of Argonne National Laboratory from 1950 to 1953. He recently joined the faculty of Stanford University in the Department of Mechanical Engineering.

I. HEAT TRANSFER

During the past ten years a great deal of effort has been directed toward obtaining a better understanding of boiling heat transfer (33).* With high-performance machines such as rockets, jet engines and nuclear reactors, tremendous quantities of heat are generated in comparatively small volumes. The heat removal problem in such devices may be illustrated by comparison with the furnace space of a modern steam boiler, in which a heat release of 40,000 Btu/(hr)(cu ft) is considered good practice. In a jet engine this figure is 10,000,000 B/(hr)(cu ft), while in a rocket or a nuclear reactor it may be 1,000,000,000 B/(hr)(cu ft). One method of removing this heat or of cooling the confining structure is by utilizing boiling with its high heat transfer rates and almost constant heat transfer surface temperatures. Boiling heat fluxes as high as 10,000,000 B/(hr)(sq ft) have been obtained experimentally (14), and practical designs are feasible up to 2,000,000 B/(hr)(sq ft).

*Numbers in parentheses refer to similarly numbered citations in bibliography.

These heat fluxes can be contrasted to that in a modern boiler, about 100,000 B/(hr) (sq ft).

It is characteristic of nuclear reactors and, to a large extent, of jet engines and rockets that they are constant heat input systems. That is, regardless of the cooling rate, a constant quantity of heat is generated, and if the cooling is inadequate the surface transferring heat will fail either by melting or possibly by very rapid corrosion caused by its high temperature. If the cooling were to have been stopped in the experimental apparatus mentioned above, operating at 10,000,000 B/(hr) (sq ft), the metal would have melted in a fraction of a second.

The design of nuclear reactors represents an unusual engineering problem because of the comparative lack of information from small-scale models and from operating experience with closely related devices. This difficulty may be partly attributed to the military necessity of immediate development of this new energy source on a large scale and to its basic differences from conventional energy sources, but the one obvious characteristic of nuclear reactors which precludes inexpensive, small-scale experimentation is the requirement of the critical mass. The recent rapid development of these inherently high-cost devices, together with that of jet engines and rockets, has been instrumental in furthering much of the recent research in the field of boiling. It is the purpose of this paper to discuss recent developments in heat transfer with boiling liquids, with emphasis on work related to the high-performance systems just mentioned. A succeeding paper (Part II. Pressure Drop) will discuss pressure drop variation and flow behavior peculiar to flow systems in which boiling occurs. No attempt will be made here to repeat the thorough discussion by Jakob (17) on the mechanism of boiling nor the very complete review by McAdams (23) of boiling research to the year 1951.

COMPLEXITY OF THE PROBLEM

Boiling heat transfer involves a change in phase from liquid to vapor. Where the viscosity, density, thermal conductivity and specific heat of the fluid could be used to describe single phase heat transfer, in boiling many additional properties are required. The surface tension, latent heat of vaporization, saturation temperature, density of vapor, and, possibly, other properties of both the liquid and the vapor must now be considered. In addition to the physical properties of both phases, the geometry and mass flow rate must also be considered, as well as the character of the surface which is transferring heat. The type of metal, the surface roughness and the presence of adsorbed gas have all been found to affect heat transfer. Usually, because of the large number of variables, a separate investigation has been required for each combination of fluid and surface. When the conditions are unusual or extreme, as is frequently the case with the high-performance devices just discussed, the cost of investigation is great. With the generalizations which are resulting from current research, however, it seems likely that the time is near when these separate investigations will no longer be necessary.

In most boiling investigations the determination of the heat transfer coefficient is a minor part of the information which is required. Data which may be of greater importance are the maximum heat that can be transferred by

the surface without overheating; the pressure drop; the stability of the boiling and of the flow; the density of the liquid-vapor mixture; and other information. The mixture density is especially important in natural circulation systems, where it controls the flow rate, and in nuclear reactors, where the reactivity is a function of the moderator density.

POOL BOILING AND NATURAL CONVECTION

Consider a submerged heating surface in a saturated liquid of practically infinite extent. As the surface temperature is increased to exceed the saturation temperature by a small amount, heat is transferred to the liquid at the heated surface, superheating the liquid. This superheated liquid may then rise to the liquid surface by convection, and evaporation will take place at the surface. This process, although it involves evaporation, is merely natural convection cooling of the heater, since only a single phase is present at that surface.

As the surface temperature increases further, vapor bubbles may form at favored places ("nuclei") on the heater surface and nucleate boiling begins. Most of the heat is still transferred from the surface to the liquid, however, with the bubbles receiving a minor fraction of the heat directly from the surface. After the buoyant force on a bubble exceeds the force binding it to the surface (a surface tension effect), the bubble breaks away and rises through the liquid. The bubble growth then occurs rapidly as heat flows from the surrounding superheated liquid and vaporizes water from the film surrounding the bubble.

Throughout the process just described, the heat flux, q″, increases rapidly with increasing surface temperature or, since saturation temperature is assumed constant, with increasing temperature difference, ΔT_x. At some critical value of ΔT_x, however, the rate of heat generation, and therefore of vaporization, becomes so great that a maximum is reached in the curve of q″ vs ΔT_x. Figure 1 shows a graph

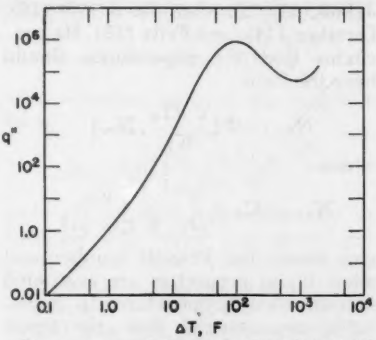


Figure 1. Typical pool boiling data (10) for water at atmospheric pressure. Heat flux is in B/(hr.)(sq. ft.).

of the data obtained by Farber and Scorah (10) for distilled water boiling at atmospheric pressure from a horizontal Chromel wire, 0.040 in. diameter and 6 in. long. At a temperature difference greater than about 100 F, the heat flux decreases, reaching a minimum at about 1000 F. If the heat transfer coefficient, h (equals $q''/\Delta T$), is plotted against ΔT_x, a curve of similar shape is obtained, but the maximum and minimum in h occur at different values of ΔT_x than those for q″. At any rate, the region of boiling in which the slope of the boiling curve is negative is termed unstable film boiling. Beyond this, the heating surface is entirely covered with vapor and stable film boiling occurs. For the high temperatures at which this type of boiling occurs with water, much of the heat is transferred by radiation.

Returning to the very important portion of the curve described as the nucleate boiling region, we shall de-

scribe a method proposed by Rohsenow (29) for the correlation of heat transfer data for surface boiling of liquids. The author offers a very plausible explanation for the mechanism of heat transfer in pool boiling, building his theory on previous observations and theories of Rohsenow and Clark (31), Jakob (17), Gunther and Kreith (15), Gunther (14), and Fritz (12). He concludes that the dependence should have the form

$$N_{Re,b}=\Phi\left(\frac{c\,\Delta T_x}{h_{fg}}, N_{Pr}\right)$$

where

$$N_{Re,b}=C_R\,\beta\frac{q''}{\mu h_{fg}}\sqrt{\frac{g_c\sigma}{g(\rho-\rho_v)}}$$

and where the Prandtl number and other liquid properties are evaluated at saturation temperature. In correlating experimental data, the terms C_R and β were omitted because of a lack of experimental observations which would permit their evaluation. The temperature difference which is used, ΔT_x, is defined as the difference between the surface temperature and the equilibrium saturation temperature at system pressure, and is usually used in boiling heat transfer correlations instead of the surface temperature minus the bulk fluid temperature, ΔT, which is used in non-boiling studies. In this way, the exact magnitude of liquid superheating or subcooling need not be known; numerous experiments have shown the validity of this method.

Rohsenow correlated a large amount of data for pool boiling of water and other liquids, but the most pertinent to the present discussion is his treatment of the data of Addoms (1), for which he obtained

$$\frac{c\,\Delta T_x}{h_{fg}}=a\left(\frac{q''}{\mu h_{fg}}\sqrt{\frac{g_c\sigma}{g(\rho-\rho_v)}}\right)^{0.33}(N_{Pr})^{1.7}$$

where the spread of the data was approximately ± 20 per cent with a value of 0.013 for the constant, a. Figure 2 shows a graph of this equation, as well as the original data of Addoms which were obtained with a platinum wire 0.024 in. diameter at water pressures from atmospheric to 2465 psia.

As Rohsenow indicates, other variables influence nucleate boiling heat transfer, and the coefficient on the right-hand side of the equation, a, is empirical and will vary with different fluid and surface combinations. Probably the most important variable not included in his correlation equation is the bubble contact angle, β. Figure 3

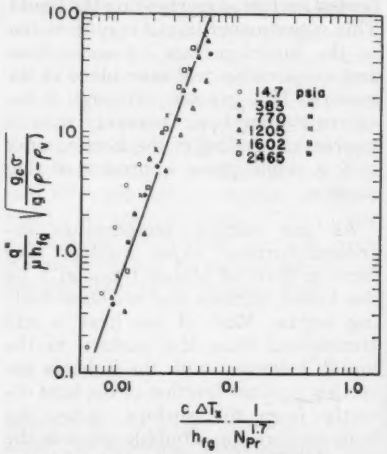
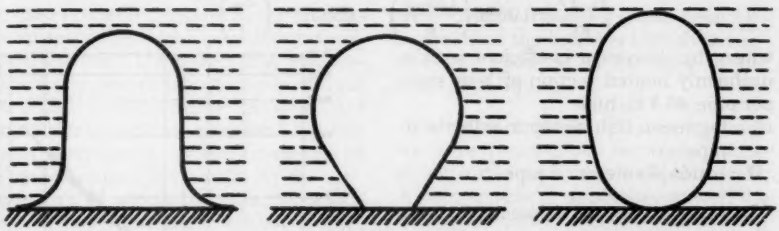


Figure 2. Correlation by Rohsenow (29) of data of Addoms (1) for pool boiling of water on a 0.024 in. diameter platinum wire.

shows β, which is a measure of the degree of wettability of a surface with a particular fluid, as observed by Jakob (17). It is seen that the contact angle decreases with greater wetting, with the result that the heat transfer rate increases for a given temperature difference, ΔT_x.

The effect of surface wetting on the heat transfer coefficient is well illustrated by some recent work with boiling liquid metals (22). Figure 4 shows these results for the various liquid metals which were used. Sodium,



(a) Not wetted (b) Partially wetted (c) Totally wetted

Figure 3. The effect of wetting property on bubble shape (17).

sodium-potassium and mercury with 0.02 per cent magnesium and 0.001 per cent titanium show the familiar sharp increase of heat transfer coefficient with increasing temperature difference in the boiling region, which is characteristic of nucleate boiling. However, cadmium, pure mercury and mercury with 0.1 per cent sodium added are non-wetting and apparently do not undergo nucleate boiling, but go directly from the forced convection region into the film boiling region.

The theoretical equation which was derived by Bromley (3) for pool boiling in the stable film region has been applied to experiments in which liquid oxygen boiled outside single horizontal tubes by Banchero et al. (2). Bromley's equation, without the radiation term, is

$$h_{co}=0.62\left[\frac{k_v^3\,\rho_v\,(\rho-\rho_v)}{D\mu_v\Delta T}\right]^{1/4}$$

and was verified experimentally by Bromley with water and several organic liquids, with tube diameters of 0.188 to 0.35 in. Banchero et al. varied ΔT_x from 5 to 500°F, pressure from 5 to 500 psia and tube diameter from 0.025 to 0.750 in. The Bromley correlation was found to predict the effects of ΔT_x and pressure but did not predict correctly the effect of diameter over the entire range. Other experiments (24) with water using wires of 0.004 to 0.024 in. diameter also indicated that the theory was inadequate at small diameters. It appears that some modification will be necessary to explain the diameter effect.

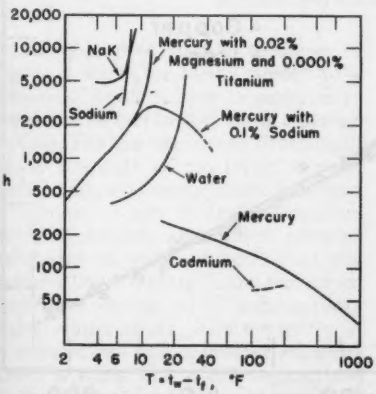


Figure 4. Comparison of experimental boiling heat transfer coefficients for water and liquid metals (22).

In the preceding discussion of heat transfer in pool boiling, the fluid flow was caused entirely by natural convection. Another important case of natural convection occurs in vertical heated tubes in which liquid is vaporized, combined with a liquid-vapor separator at the top and with unheated downcomers to bring liquid to the bottom of the heated tubes. A recent paper by Piret and Isbin (28) presents heat transfer data for a single heated tube in a system of this type, correlated by two different methods. An empirical relation which was offered for water and five other fluids at atmospheric pressure was the following:

$$\frac{h_{av}D}{k}=0.0086\left(\frac{Du_m\rho}{\mu}\right)^{0.8}(N_{Pr})^{0.6}\left(\frac{\sigma_w}{\sigma}\right)^{0.33}$$

where h_{av}=average coefficient over a uniformly heated section of 1-in. copper pipe, 46.5 in. high

u_m=log mean liquid-vapor velocity in pipe

D=inside diameter of pipe

$\frac{\sigma_w}{\sigma}$=ratio of surface tension of water to that of the fluid used, both evaluated at their normal boiling points

Figure 5 shows a graph of

$$\left(\frac{h_{av}D}{k}\right)(N_{Pr})^{0.6}\text{ vs. }\left(\frac{Du_m\rho}{\mu}\right)^{0.8}\left(\frac{\sigma_w}{\sigma}\right)^{0.33}$$

together with the data presented by the authors. The other correlation which was used will be discussed below in connection with Rohsenow's extension of his correlation method to forced convection boiling.

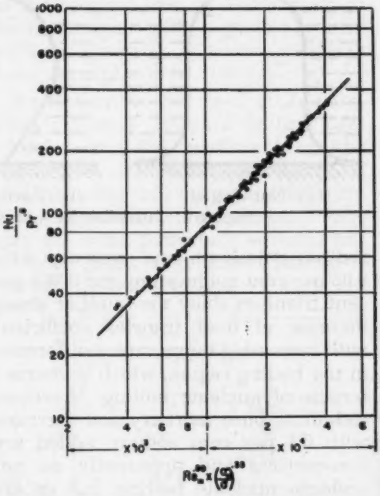


Figure 5. Correlation of heat transfer data for six fluids in 1-in. diameter vertical-tube natural circulation evaporator (28). A log mean velocity is used in the Reynolds number, Re_m. Nu and Pr denote Nusselt number and Prandtl number of saturated liquid.

Returning to the pool boiling model discussed previously and to the boiling curves in Figure 1, we should point out that the maximum heat flux is in-

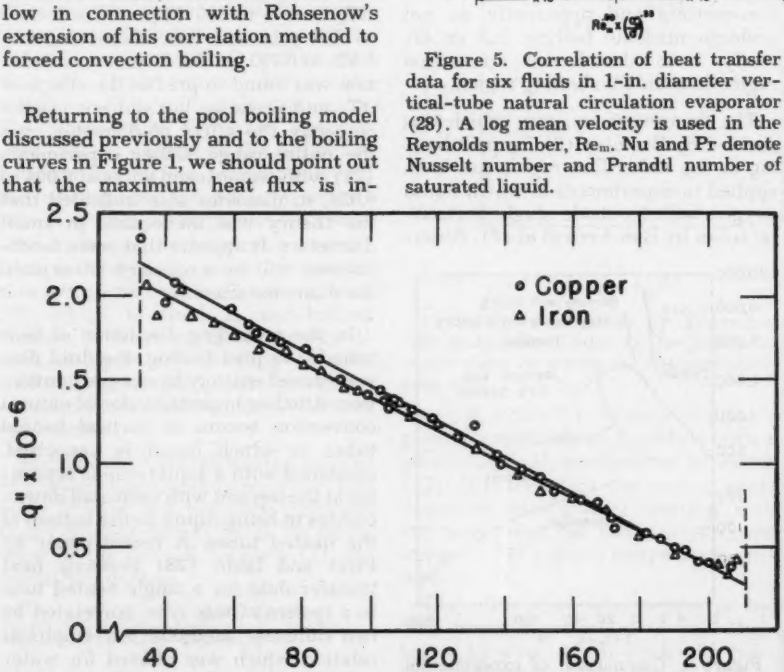


Figure 6. Maximum heat flux for copper and iron wires with pool boiling of water at atmospheric pressure (11). Broken lines show freezing and boiling temperatures.

creased considerably if the bulk temperature of the liquid is below saturation. Farber (11) found that for horizontal copper or iron wires immersed in water at atmospheric pressure, the maximum heat flux increased linearly with decreasing water temperature. Figure 6 shows the results for the copper wire in which the maximum heat flux varied from about 450,000 B/(hr) (sq ft) with the bulk temperature at 212°F to about 2,200,000B/(hr) (sq ft) at 32°F (180°F subcooling).

It is evident that the nature of the boiling process is somewhat different when the bulk of the liquid remains subcooled. With moderate rates of heat input, the heater will be cooled adequately by convection, and no vapor will be formed. If the heat flux is increased, however, the surface temperature will increase until liquid near the surface becomes sufficiently superheated to vaporize, and the phenomenon known as local boiling will occur. Under conditions of moderate flux and relatively large subcooling, the vapor bubbles grow and collapse without seeming to leave the surface, but with lower subcooling it has been observed that the bubbles leave the surface and collapse in the bulk of the subcooled liquid.

In spite of these differences which we have pointed out between boiling in saturated and in subcooled liquids, it is important to emphasize as well the similarities between the two. For most practical purposes the relationship between q'' and ΔT_x is the same for the two cases, and, in local boiling, ΔT_x is nearly independent of the amount of subcooling. The values of q''_{max} are definitely greater for subcooled boiling, however.

Because of the higher maximum heat flux attainable, local boiling is quite important to high-performance applications. Data are lacking from the literature on local boiling heat transfer in natural circulation systems such as the one described in connection with the work of Piret and Isbin (28), but quite a lot of experimentation has been conducted recently in forced circulation systems. This will be discussed below.

BOILING WITH FORCED CONVECTION

Boiling curves similar to those in Figure 1 exist for heat transfer to a saturated liquid which is pumped or otherwise forced to flow along a heated surface. For the important situation in which the bulk of the liquid is subcooled (i.e., at a temperature less than saturation) when it first encounters the heat source, and which remains subcooled in spite of boiling at the heater ("local boiling"), the maximum heat flux occurs at a substantially higher value of ΔT_x. Consequently, it is difficult to find a heater material which will not fail in this region, and the complete curve has not yet been experimentally verified for forced convection local boiling.

A number of experiments (18,25,32) have shown that boiling heat transfer coefficients are not significantly dependent on the fluid velocity or the fluid temperature, but that they do depend on the pressure and the temperature difference between the surface and saturation (ΔT_x). Jens and Lottes (18) correlated data on local boiling of water which flowed vertically upward in electrically heated tubes of stainless steel or nickel, having inside diameters ranging from 0.143 to 0.226 in. and L/D ratios from 21 to 168. Pressures ranged from 85 to 2500 psia, bulk water temperatures from 229 to 636°F, mass velocities from 8,000 to 7,700,000 $lb_m/(hr)(sq\ ft)$ (corresponding to

liquid velocities of 0.04 to 40 fps) and q″ up to 4,000,000 B/(hr)(sq ft). Their recommended correlation is

$$\Delta T_x = \frac{60(q''/10^6)^{1/4}}{e^{p/900}}$$

where p is the absolute pressure in psia and q″ is in B/(hr)(sq ft).

Rohsenow (30) has extended his correlation for pool boiling to forced convection surface boiling by considering the total heat flux to be made up of two components, one a "boiling flux" and the other a "convective flux."

$$q'' = q_b'' + q_c''$$

In order to apply this correlation, one first calculates the convective flux by means of the conventional Dittus-Boelter equation for adiabatic, single-phase flow, modified by using a coefficient of 0.019 instead of 0.023 (30):

$$q_c'' = h_c(t_w - t_f)$$

$$\frac{h_c}{cG} N_{Pr}^{2/3} = 0.019 \left(\frac{GD}{\mu}\right)^{-0.2}$$

The boiling flux, q_b'', may then be computed by difference, using total flux (q″) and q_c''. This value of q_b'' is used in his previously-obtained pool boiling correlation which has already been discussed.

$$\frac{c\,\Delta T_x}{h_{fg}} = a\left(\frac{q_b''}{\mu h_{fg}}\sqrt{\frac{g_c\sigma}{g(\rho-\rho_v)}}\right)^{0.33} (N_{Pr})^{1.7}$$

where the value of the constant, a, varies with various liquid-surface combinations (Table 1).

Application of this theory has been made with the natural circulation evaporator system described previously. Reference has already been made to one correlation which Piret and Isbin (28) presented for their data from a single heated tube. They also correlated their data using Rohsenow's dimensionless groups with the flux q''_b calculated as described above, with the results shown in Figure 7. For comparison, the pool boiling correlation obtained by Rohsenow (30) from the data of Addoms (1) for water is also plotted on this graph. It is seen that the agreement of the data is excellent for water and that all the data can be correlated by selecting appropriate values of the coefficient, a, corresponding to the particular fluid-surface combination, as Rohsenow predicted. Table 1, reproduced from reference 28, presents values of this coefficient for various liquid-surface combinations.

TABLE 1.

Values of the Coefficient, a, for Various Liquid-Surface Combinations

Combination	a
Water-copper (28)	0.013
Carbon tetrachloride-copper (28)	0.013
35% K_2CO_3-copper (28)	0.0054
n-Butyl alcohol-copper (28)	0.0030
50% K_2CO_3-copper (28)	0.0028
iso-Propyl alcohol-copper (28)	0.0022
n-Pentane-chromium (30)	0.015
Water-platinum (30)	0.013
Benzene-chromium (30)	0.010
Water-brass (30)	0.0060
Ethyl alcohol-chromium (30)	0.0027

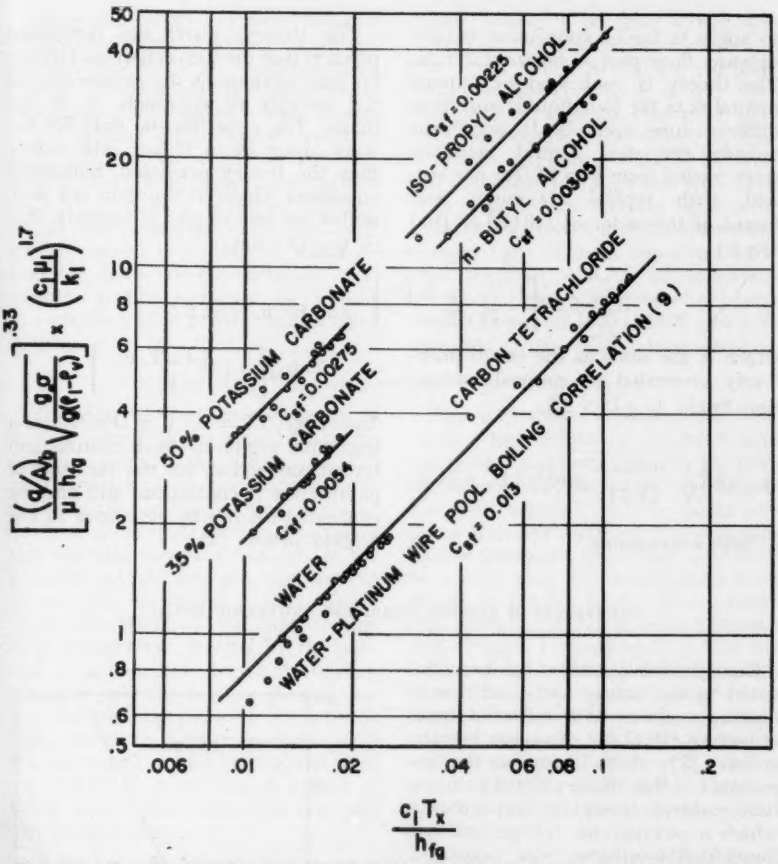


Figure 7. Correlation of boiling heat transfer data of Figure 5 by method of Rohsenow (29,30). Subscript "1" refers to saturated liquid state.

Practically nothing has been reported on unstable film boiling with forced convection, probably as much because of a lack of apparent application of such data as because of the difficulty of experimentation in this region with common fluids. At least one exception exists, however, since a recent paper (26) describes a method of boiling liquid oxygen and nitrogen which are pumped through tubes heated by air. Because of the large temperature difference, all the measurements were in the unstable region between nucleate and film boiling. The influences of temperature difference, pressure and flow rate were examined, the pressure being supercritical in some of the tests. The boiling heat transfer coefficient was found to decrease with increasing temperature difference in all cases where boiling took place.

Bromley's theoretical equation for stable film boiling has been extended

to apply to forced convection (4) for upward flow past a horizontal tube. The theory is confirmed by experimental data for four liquids and three different tube sizes: ⅜, ½, and ⅝-in. outside diameter. Liquid velocities were varied from 0 to 14 feet per second, with typical maximum heat fluxes of the order of 140,000 B/(hr)(sq ft).

$$h_{co}=0.62\left[\frac{k_v^3(\rho-\rho_v)\rho_v g\lambda'}{D\,\Delta T_x \mu_v}\right]^{1/4} \quad \text{and} \quad h=h_{co}+\tfrac{3}{4}\,h_r$$

which is the same as the result previously presented for natural convection. For $u/\sqrt{gD} > 2.0$,

$$h_{co}=2.7\sqrt{\frac{u\,k_v\rho_v\lambda'}{D\,\Delta T_x}} \quad \text{and} \quad h=h_{co}+\tfrac{7}{8}\,h_r$$

In both expressions,

$$\lambda'=h_{fg}\left[1+\frac{0.4\,\Delta T_x\,c_{pv}}{h_{fg}}\right]^2$$

The theory which was developed predicts that the convection heat transfer will increase as the square root of the velocity of approach, u, of the liquid. The experimental data for h_{co} were about 20 to 40 per cent higher than the theory predicted. Simplified equations which fit the data are presented for two ranges of velocity. For $u/\sqrt{gD} < 1.0$,

The range, $1.0 \leq u/\sqrt{gD} \leq 2.0$, is a transition region between natural and forced convection, but the variation of parameters is continuous and may be evaluated readily by reference to the original article (4).

THE EFFECTS OF SURFACE CONDITION AND DISSOLVED GAS

Considerable attention has been devoted to the nature and condition of heating surfaces with different types of boiling (10,17,23). A recent investigation (32), which illustrates the importance of this effect, utilized a single-tube, natural circulation evaporator in which a commercial refrigerant, trifluorotrichloroethane, was vaporized. The heat transfer surface consisted of the interior of an electrically heated copper tube, 0.625-in. inside diameter and 80 in. long. The surface condition was classified in various tests as either "mechanically polished," "acid-cleaned," or "overheated." Figure 8 shows the trend of the data from these tests, from which may be seen the very great effect of changing surface condition. The "overheated" classification came about when the surface temperature was accidentally allowed to exceed 600°F, after which it was observed that the heat transfer results were quite changed.

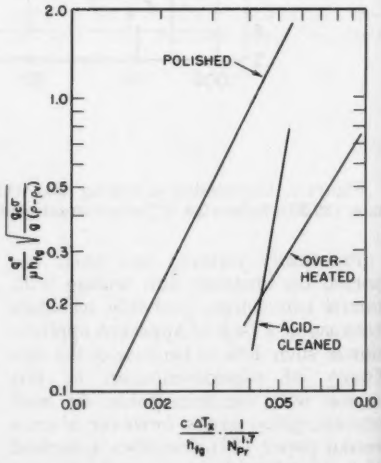


Figure 8. Nucleate boiling data showing effect of different surface conditions (32).

Corty and Foust (7) have found a pronounced effect of surface roughness in experiments with boiling ether, normal pentane and trifluorotrichloroethane. Profilometer measurements of the heating surfaces were made, as well as photo-and electron-micrographs, and photographs were made of the rising columns of vapor bubbles. These results and other similar work go far toward providing an explanation for some of the different heat transfer results obtained by earlier investigators with superficially similar systems.

The presence of gas, either adsorbed on the surface or dissolved in the liquid, tends to initiate nucleate boiling at lower values of ΔT_x than with gas-free systems. Pike et al. (27) have investigated the effect of dissolved air and carbon dioxide in water and glycerine boiling from wire coils at low pressures and, in agreement with others, have found the amount of dissolved gas to be an important variable. Buchberg et al. (5) found a similar though much smaller effect with distilled water at 1000 to 3000 psia with up to 900 cc/liter of dissolved nitrogen.

CONSTANT HEAT INPUT VS. CONSTANT TEMPERATURE HEATING

An important characteristic of a boiling system is the type of heating which is used. If the heat input will continue at a given rate regardless of the condition of the cooling medium, we have a constant heat input system and say that the heat flux, q″, is the independent variable. An ideal system of this type would be a wire whose electrical resistance did not change with temperature, heated by the passage of a constant electrical current. We shall add the further idealization that the melting point of the wire is high enough to permit operation well into the range of film boiling and then use this wire, submerged in a pool of liquid, as a model to describe constant heat input boiling.

Figure 9a shows a simplified boiling curve which might be obtained with the system just described. By increasing the current step-wise, starting at the point labelled "A", we could obtain a series of values of ΔT, as shown by the graph. At each step, of course, we would wait for steady-state conditions to prevail before taking readings. Between J and B, prior information (Figure 1) would tell us that the boiling curve is multivalued, with three different values of the dependent variable, ΔT, for each value of q″. However, operation is stable throughout the region $0 \leq q'' < q''_{max}$, and it is not until q″ exceeds q''_{max} that the remainder of the curve becomes important.

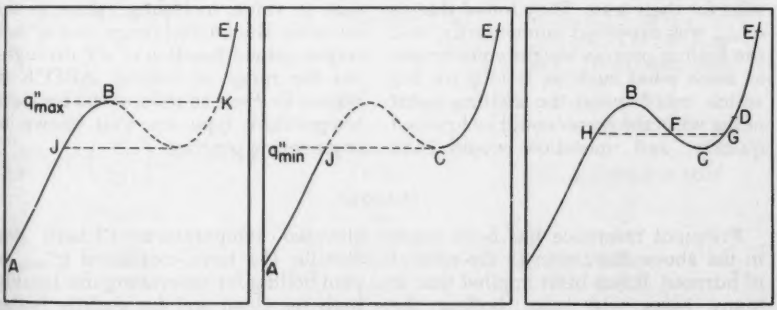


Figure 9. Typical boiling curves (q″ vs ΔT_x) similar to Figure 1. These illustrate types of heating.

As soon as q''_{max} is exceeded, the only possible operation is in the film boiling region, point K and beyond. The "jump" from B to K entails sudden transition from nucleate to stable film boiling, together with a large increase in ΔT. With many real systems, this results in a heater temperature greater than the melting point, and the surface is destroyed. With the ideal system postulated, however, the discontinuous curve ABKE describes boiling operation when the heat flux is increased in steps from near zero to above q''_{max}.

If, now, the heated wire is operating in the stable film boiling region, say at E in Figure 9b, and the heat flux is decreased in a series of steps, the operation will be along the curve ECJA. There is also a discontinuity in this curve, from C to J, but the transition is from film to nucleate boiling, with an accompanying drop in ΔT, and would have no harmful result except, perhaps, mechanical damage from thermal shock.

The discussion to this point would indicate that a constant heat input system may not be expected to operate in the unstable film boiling region. The reader may be wondering, therefore, how the data in Figure 1 were obtained, since it was stated that Farber and Scorah (10) used an electrically heated wire for their tests. They found that as q''_{max} was exceeded momentarily, and the boiling process sought equilibrium at some point such as E (Figure 9c) which was beyond the melting point of the wire, the power could be lowered quickly, and operation would take place at some point such as D. This permitted operation in the range from C to D. They further found that, starting at C, a slowly applied increase of power sometimes resulted in operation between C and B. Starting at F in the unstable region, a sudden power increase caused a shift to some point G in the film boiling region, while a sudden decrease caused a shift to point H in the nucleate boiling region.

Electrical resistance heaters, as well as nuclear reactors, rockets and jet engines, represent essentially constant heat input systems. Strictly speaking, of course, electrical resistance will vary with temperature, nuclear reactivity will vary with moderator density, etc., so that the discontinuous jump to film boiling (and usually burnout) does not occur at exactly constant heat flux.

A constant temperature system is typified by a heat exchanger in which heat is transferred from a liquid metal to boiling water. To idealize the system, assume that the flow of liquid metal is so great that its temperature drop in the heat exchanger is negligible. Assume, further, that the liquid metal temperature may be regulated by external means to any desired value and that the thermal resistances of the liquid metal film and of the tube are negligible. We may now set ΔT at any desired value, including values in the unstable film boiling range, and q'' is a single-valued function of ΔT throughout the range of boiling, ABFCE in Figure 9c. Heat sources of the constant temperature type are well known in engineering practice.

BURNOUT

Frequent reference has been made in the above discussion to the subject of burnout. It has been implied that in many cases with pool boiling the heater will not fail if q''_{max} is exceeded, but will continue to operate at an elevated temperature. Cichelli and Bonilla (6) have correlated q''_{max} in pool boiling for several organic liquids, both for clean and for slightly fouled surfaces. Other investigators have obtained data for water in agreement

with this correlation, which predicts that q''_{max} is greatest (for water) at about 1000 psia, the heat flux here being approximately 1,250,000 B/(hr)(sq ft).

For local boiling with forced circulation the attainable heat fluxes are much higher than for pool boiling. Table 2 shows a summary of recent correlations which have been proposed to predict burnout in local boiling. Generally, experimental data were obtained in a long heated tube, and burnout occurred (or overheating was detected) at the outlet end. All the correlations are seen to be based on the subcooling, $\Delta T_{sub} = t_{sat} - t_f$, at the exit of the tube, where the bulk fluid temperature t_f is a calculated value based on flow rate, heat input and inlet temperature. In the last correlation of Table 2 both the constant C and the exponent m were found to be functions of pressure.

It would appear from most of the correlations in the table that only the

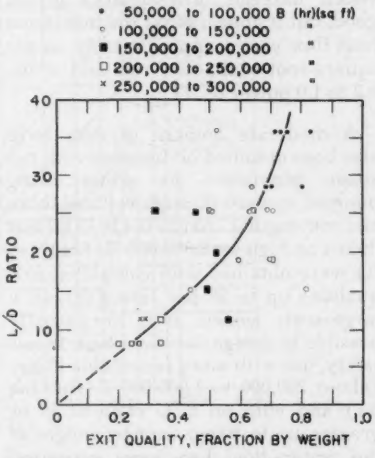


Figure 10. The effect of L/D ratio on exit quality at burnout for a 0.94 in. vertical tube with upflow of water. Exit pressure, 15 psia; inlet subcooling, 150F; and inlet velocity, 0.02 to 0.2 fps (19).

TABLE 2.

Forced Convection Burnout Correlations

Formula	*Source*	*Range of Variables*
$q''_{max} = (400{,}000 + 4800\ \Delta T_{sub})u^{1/2}$	McAdams (23)	u from 1 to 20 fps p from 30 to 0 psia ΔT_{sub} from 20 to 100F
$q''_{max} = 7000\ u^{1/2}\ \Delta T_{sub}$	Gunther (15)	u from 5 to 40 fps p from 14 to 160 psia ΔT_{sub} from 20 to 280F
$q''_{max} = 520\ G^{1/2}\ \Delta T_{sub}^{0.22}$	Buchberg et al. (5)	u from 5 to 30 fps p from 500 to 2000 psia
$\frac{q''_{max}}{10^6} = C\left(\frac{G}{10^6}\right)\Delta T_{sub}^{0.22}$	Jens and Lottes (18)	ΔT_{sub} from 3 to 160F

P(psia)	m	C
500	0.16	0.817
1000	0.28	0.626
2000	0.50	0.445

Note: q''_{max} is expressed in B/(hr)(sq ft)
G is expressed in lb_m/(hr)(sq ft)
u is expressed in fps

two variables of subcooling and flow rate affected burnout. Agreement between different investigators is not good, but it appears that the maximum heat flux varies approximately as the square root of the flow rate and as the 0.2 to 1.0 power of ΔT_{sub}.

A moderate amount of data have also been obtained on burnout with net steam generation for water being pumped upward through vertical tubes and rectangular channels (18,19). Heat fluxes as high as 3,000,000 B/(hr) (sq ft) were obtained with calculated exit qualities up to 30 per cent (18). In a large-scale system it is not usually possible to design for these high fluxes safely, but with more reasonable fluxes (about 200,000 to 1,000,000 B/(hr) (sq ft)) and with an L/D ratio of 50 or greater, up to 80 per cent by weight of the water flow has been vaporized without serious increase in surface temperature. Figure 10 indicates the effect of L/D ratio for a vertical heated passage, from which it is seen that as L/D is decreased, the exit quality at which overheating occurs also decreases. In very short tubes the central core of liquid probably passes the heat transfer surface before mixing can occur, so that a considerable fraction of the liquid is not available for vaporization, and the tube overheats at relatively low calculated mixture qualities.

It would be expected that, for horizontal tubes or surfaces, flow separation and consequent overheating might occur at very low qualities. In one series of tests, however, as much as 44 per cent by weight of water was vaporized with heat fluxes up to 350,000 B/(hr) (sq ft) without overheating (21). The L/D ratio in these tests was about 128, the flow rates were from 110 to 360 lb_m/(sec) (sq ft), and the outlet pressure from 30 to 300 psia.

No correlation has yet been proposed for burnout with forced convection and net steam generation in horizontal tubes, and, although data have been obtained (18,19) for vertical and inclined tubes, no general correlation has been made. It is the opinion of the authors that the variation in flow characteristics of the experimental systems used by various investigators have contributed greatly to the variance in results. Because burnout does not occur until a considerable volume of vapor is being generated, there is an accompanying sharp increase in pressure gradient near the outlet of the tube. Depending upon the characteristics of the circulation system, the result may be a decrease in flow rate which, with constant heat input, results in more vapor generation and rapid overheating of the surface. This complicating effect of pressure drop variation will be discussed in the second paper of this series.

BOILING AND BURNOUT WITH TWO COMPONENTS

Another item of significance in the utilization of boiling heat transfer is the boiling of a mixture of two miscible or immiscible components. It is possible that burnout in a system might be prevented by the addition of a second fluid which has a slightly higher saturation temperature. Figure 11 shows the heat transfer behavior of a mixture of hydrocarbon fuels (16). The expected increase in heat transfer occurs when the surface temperature reaches a definite amount, as shown by the sharp change of the curves to near-vertical. At a higher heat flux, another sharp change in the slope occurs, and the curves become nearly horizontal. If continued, this trend would lead to burnout with almost no further increase in flux. Instead, however, the temperature of the surface seems to increase only to a value associated with the original heat transfer coefficient and would probably not cause destruction of the surface. No application of this principle to prevent burnout in high-performance equipment has yet been announced.

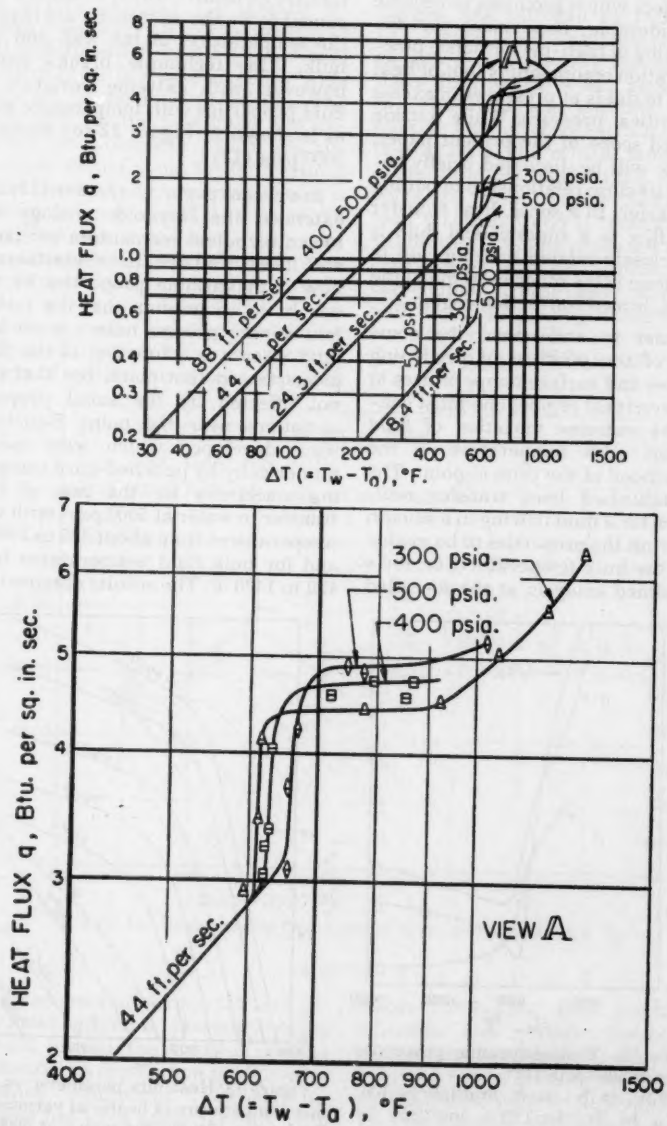


Figure 11. Heat transfer behavior of a mixture of hydrocarbon fuels (16). T_s refers to bulk fluid temperature.

A subject which promises to become of considerable importance in the engineering of high-performance power generation equipment is that of heat transfer to fluids at pressures in excess of the critical pressure. While outside the stated scope of the present paper, this topic will be discussed briefly because of its close relationship to boiling heat transfer. In a sense, heat transfer at high flux to a supercritical fluid is a more closely related limiting case to boiling than is the other extreme in the low-flux, forced convection region.

In order to understand the complexity of the problem of predicting heat rates and surface temperatures in the supercritical region, one must consider the extreme variation of fluid properties with temperature in the neighborhood of the critical point. The well-established heat transfer relationships for a fluid flowing in a smooth tube permit the properties to be evaluated at the bulk temperature, or, for a more refined analysis, at the so-called film temperature, which is usually assumed to be the arithmetic average of the temperatures at the wall and the bulk. This technique breaks down, however, with extreme variation of fluid properties with temperature such as is shown in Figure 12 for water at 5000 psia (13).

In a recent paper, Goldmann (13) has extended the Reynolds analogy between turbulent momentum exchange and heat transfer to nonisothermal flow with variable properties by the one basic assumption that the turbulent mixing process near a point in a fluid stream is a function of the fluid properties at that point, but that it is not affected by the small property variations near the point. Equations were developed which were solved numerically by punched-card computing machinery for the case of heat transfer to water at 5000 psia, with wall temperatures from about 425 to 1400°F and for bulk fluid temperatures from 450 to 1400°F. The results obtained are

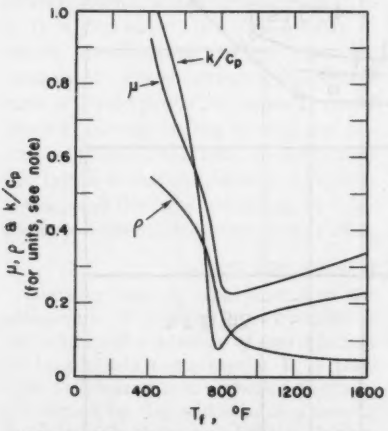


Figure 12. Thermodynamic properties of water at 5000 psia (13).

Note: For ρ in lb.$_m$/cu ft., multiply by 100
For μ in lb./(sec.)(ft.), multiply by 10,000
For k/c_p in lb./(sec.)(ft.), multiply by 10,000

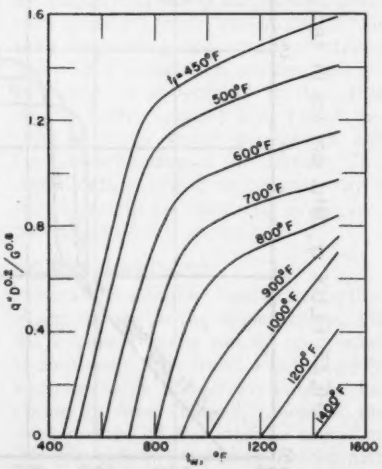


Figure 13. Heat flux parameter vs. surface temperature of heater at various bulk fluid temperatures for water at 5000 psia (13). (q" in B/(sec.)(sq. ft.), D in ft. and G in lb.$_m$/(sec.)(sq. ft.).

reproduced in Figure 13. In addition, the author checked his method by comparison with Deissler's analytical and experimental results (8) for heat transfer to air in a case involving wide variation of the fluid properties. The agreement for the case cited was excellent. As Goldmann points out, the designer should avoid the flat portions of the curves if surface temperature is a limiting factor, since it is there that a small increase in the heat input produces a relatively large increase in surface temperature.

Heat transfer measurements for oxygen and nitrogen at pressures in excess of critical have been reported by Monroe et al. (26) and for hydrogen, oxygen, and nitrogen by Dunn et al. (9). Figure 14 shows typical results for oxygen as reported by the latter authors, from which it can be seen that the film coefficient exhibits a pronounced and unexpected minimum in the region of the critical temperature of the fluid. At higher fluid temperatures the heat transfer coefficient is nearly constant and is consistent with values that could be predicted from the conventional convective heat transfer correlations. These tests were made with the fluid passing through a long electrically heated tube, with heat fluxes as high as 4,660,000 B/(hr) (sq ft).

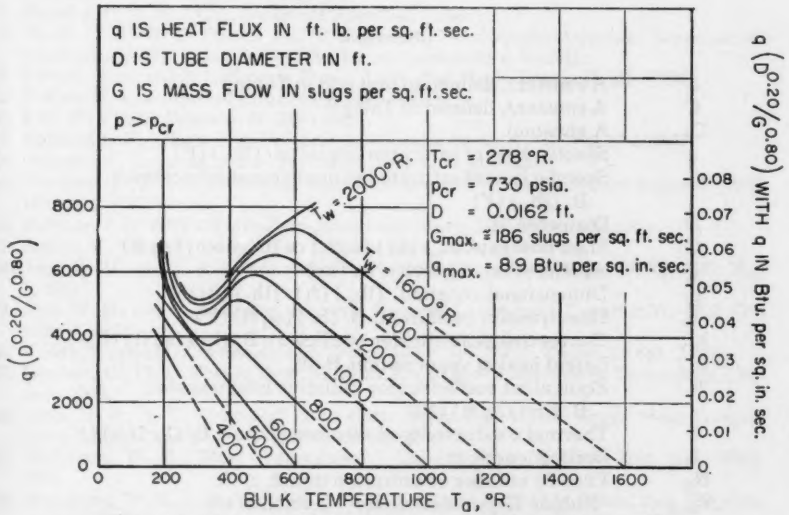


Figure 14. Heat transfer to oxygen at supercritical pressure (9).

SUMMARY

A great amount of research and development effort is presently being directed toward the study of boiling and toward its application to high-performance equipment. Considerable progress has been made recently with improved theoretical explanations of the mechanism of both nucleate and stable film boiling. Heat transfer coefficients and surface temperatures can be predicted with reasonable accuracy for most present applications in constant heat input systems. This is not true for constant temperature systems, but in practice the scale factor is usually so high with these systems that

an accurate determination of the boiling coefficient is unnecessary.

The burnout heat flux during local boiling with forced convection is exceedingly high for liquids that wet the heating surface. No data have been reported for non-confining surfaces, such as for boiling outside tube bundles, but with this exception, further extensive burnout investigations in the local boiling region are not urgently needed. It is to be hoped, however, that fundamental work leading to a basic understanding of the heat transfer and flow mechanism will be continued, because existing correlations are not in particularly good agreement.

Burnout data with net steam generation and forced convection are available, but no adequate general correlation has been proposed. Neither has a basic explanation of the exceedingly complex heat transfer and flow mechanism been offered. With inclined or horizontal tubes and with natural convection systems, the situation is much worse, since almost no information is available on burnout in these cases. One factor which is obviously of paramount importance to the subject of maximum safe heat flux in these flow systems is the variation of pressure drop during the flow of a boiling fluid. This subject will be discussed in Part II of this paper.

NOTATION

a	A constant, defined in text and in Table 1
C	A constant, defined in Table 2
C_R	A constant
c	Specific heat of saturation liquid, B/ (lb_m) (F)
c_{pv}	Specific heat of saturated vapor at constant pressure, B/ (lb_m) (F)
D	Diameter, ft
G	Mass flow rate, lb_m/ (hr) (sq ft) or $lb/_m$ (sec) (sq ft)
g	Acceleration of gravity, ft/ $(hr)^2$
g_c	Dimensional constant, (lb_m) (ft) / (lb_f) $(hr)^2$
h	Heat transfer coefficient, B/ (hr) (sq ft) (F)
h_{co}	Convective heat transfer coefficient, B/ (hr) (sq ft) (F)
h_{fg}	Latent heat of vaporization, B/lb_m
h_r	Equivalent coefficient for radiative heat transfer, B/ (hr) (sq ft) (F)
k	Thermal conductivity of saturated liquid, B/ (hr) (ft) (F)
L	Boiling length, ft
N_{Pr}	Prandtl number of saturated liquid, $c\mu/k$
$N_{Re,b}$	"Bubble Reynolds number," defined in text
p	Absolute system pressure, psia
q''	Heat flux, B/ (hr) (sq ft)
q''_{max}, q''_{min}	Maximum and minimum heat fluxes in boiling curves, Figures 1 and 9, B/ (hr) (sq ft)
t_f	Fluid bulk temperature, °F
t_{sat}	Saturation temperature of fluid, °F
t_w	Surface temperature of heater, °F
ΔT	Temperature difference, $(t_w - t_f)$, F
ΔT_{sub}	Subcooling, $(t_{sat} - t_f)$, F
ΔT_x	Temperature difference, $(t_w - t_{sat})$, F
u	Fluid velocity, fps

v	Subscript used with the symbols k, ρ and μ to denote the saturated vapor state
β	Contact angle of vapor bubble with heating surface
λ'	Parameter defined in text, B/lb_m
μ	Dynamic viscosity, $lb_m/(ft)(hr)$
ρ	Density of saturated liquid, $lb_m/cu\ ft$
σ	Surface tension of saturated liquid, $lb_f/ft/$
ϕ	Mathematical function of unspecified form

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TRAINING OF ENGINEER OFFICERS FOR THE ROYAL NAVY

ACKNOWLEDGEMENT

Recently, at the invitation of the British Chief of Naval Information, various British technical publications sent representatives to the Royal Naval Engineering College at Manadon to note the progress in developments of that school. This reprint combines the results of this visit, as published in the 6 August 1954 issue of *The Engineer* and the 13 August 1954 issue of *Engineering*.

HISTORY AND DEVELOPMENT

Although the Engineering Branch of the Navy has only existed for about 120 years its history is linked with the period during which the greatest changes have taken place, the transition from the ship - of - the - line of Nelson's day to the modern aircraft carrier. From Papin's engine in 1690 the steam engine has been steadily developed by Newcomen and James Watt and others and was eventually adapted to ship propulsion by men such as John Fitch, although it was not until the appearance of Henry Bell's *Comet* in 1812 that steam navigation began to be given serious consideration. The first steam vessel in the Navy was purchased into the Service in 1821, to be followed by others, but it was not until seven years later that steamships appeared in the Navy List. As the years passed the number of such ships steadily increased and the Crimean War served to demonstrate the value of steam so that a point was reached when no more ships solely relying upon sails were laid down. While steam was gaining the ascendancy over sails there was a controversy over the merits of paddles compared with the screw propeller as a means of propulsion, which culminated in the *Alecto* and *Rattler* trials which demonstrated the superiority of the screw. With the development of steam machinery the question of boiler pressures to be employed gave rise to considerable argument and 10 lb. per square inch was advocated as the safe maximum in 1846. This figure compares with the 400 lb. per square inch of the present day and serves as some measure of the advances made over the years, including the transition from the box boilers last fitted in H.M.S. *Thunderer* in 1876 to high pressure oil-fired water-tube boilers now in use.

Little is known of the earliest engineers in the Navy, the first recorded entry being in 1826, but history records that those whose duties were to tend machinery were held in little esteem and opinion was prejudiced against them. However, engineers were granted Warrant rank in 1837, after

Captain Edward Parry, the explorer, became "Comptroller of Steam Machinery" and this step is considered to be the true beginning of the Engineering Branch. Subsequent history is a record of a long struggle to achieve recognition and status, to formulate training methods and found training establishments, and to create a suitable internal organization to keep pace with technical developments. In 1841 the Engineer's Button, carrying a picture of a side-lever engine embossed, was introduced and continued, together with a special arrangement of the buttons, to act as a distinguishing mark of the Engineering Branch for fifteen years. It was then abolished although the arrangement of buttons remained until colored stripes were introduced in 1863. The division of engineer officers into three ranks, in 1847, with Inspectors of Machinery Afloat and Chief Engineers appointed by Commission and ranking with junior captains and lieutenants, respectively, marked the second step in the advancement of the branch. The two ranks mentioned above appeared in the Navy List in 1849 but it was not until 1870 that the third rank, that of Assistant Engineers, was included. In the succeeding years various new ranks were introduced, including Chief Inspector of Machinery with the rank of Rear Admiral and Fleet Engineer with the rank of Commander, and adjustments made to remove anomalies.

The Selborne-Fisher scheme came into being in 1902 to help to resolve the antagonism between the branches and eventually the Executive and Engineering Branches were amalgamated, but it was not until 1914 that most of the wrongs were redressed and peace was established.

As regards the training of engineers in the Service, in 1828 boys began to be entered as "Engineers Boys" and were trained for five years afloat. Then, for a period of three years, from 1843, "Engineers Boys" were accommodated on the barque *Sulphur*, and later as "Engineer Students" they were able to receive their education at the Royal School of Naval Architecture and Marine Engineering which was opened in South Kensington in 1864 and continued until 1873 when the Royal Naval College at Greenwich was opened. With the continued expansion of the application of steam machinery to the fleet, recruitment and training were matters requiring careful study and as a result of the recommendations of a committee, under the chairmanship of Admiral Sir Ashley Cooper-Key, the old "line of battle" ship *Marlborough* was appropriated, in 1877, as a training establishment for Engineer Students. Three years later the training school at Keyham was opened and eventually absorbed all the students when *Marlborough* was closed in 1888. Since that date Keyham has continued to train engineers for the Navy until it was closed in 1910 to be reopened in 1914. However, the first world war intervened and it was not until 1919 that the College reverted to the training of engineer officers.

In the process of time the buildings became outmoded and the fixed equipment out of date, added to which the Navy's technical demands rapidly grew, with the increased complexity and variety of machinery and auxiliaries, and there was also the need for specialist training in aeronautical and gunnery engineering. These requirements could not be met by the accommodation at Keyham, which only had space to train 150 officers under conditions of a reasonable degree of comfort and efficiency and was incapable of expansion. Under these circumstances the Admiralty purchased a 100-acre estate at Manadon, on the outskirts of Plymouth, including an early seventeenth century building, Manadon House, and an old tithe barn, for development as a training establish-

ment. A few years later, in 1940, the Royal Naval Engineering College at Manadon was opened with temporary accommodation buildings. Further temporary buildings were added and considerable sums of money were spent on equipment and buildings, including the permanent instructional block, the eastern section of which was in use by 1945. One year later both the training establishments, Keyham and Manadon, were commissioned as H.M.S. "Thunderer." Other buildings erected on the site by this time were temporary recreational block, No. 1 permanent hangar, and two pentad pattern hangars.

The constructional work mentioned above constituted what was termed the first building program. The second program consisted of the completion of the instructional block, which was accomplished in 1951, and the provision of the engineering workshops, known in the establishment as "the Factory." The instructional block now covers five acres and includes lecture rooms and laboratories for physics, chemistry, metallurgy, mechanics and hydraulics, a drawing office, and a library and reading room, stocked mainly with technical works and publications, but containing also a wide variety of other literature; for the basic characteristics of Manadon are those of a university, and not merely those of a college of technology.

The engineering workshop is notably well equipped with up-to-date machine tools, and everything necessary is also provided to teach the fundamentals of smithwork, pattern-making, etc. There is a separate workshop for instruction in aeronautical engineering, and a test-house in which jet engines can be tested. Other workshops and laboratories are allocated to ordnance engineering, including both gun mountings and gunnery control mechanisms. The liberal scale of the equipment is indicated by the fact that the engineering portion of the building contains a set of turbines from a Hunt class destroyer, and boilers on which the students practice cutting out and replacing tubes; and that in the ordnance department there is a complete mounting with twin 4-in. guns and the control gear associated with them.

At present, the electrical engineering laboratory and an engine test shop for Diesel engines, small steam engines and steam turbines are still at Keyham, but eventually these also will be transferred to Manadon. Buildings for these purposes are included in the final program necessary to complete the College, and when the transfer has been effected the existing accommodation at Keyham will be taken over by the Devonport Dockyard Technical College. Also included in this program are permanent buildings at Manadon for use in the study of aerodynamics and the structure of aircraft, and a permanent accommodation block to house a total complement of 500. This total will include the staff; the number of officers under training is about 350. It is intended, when the accommodation block becomes available, to remove the rows of huts now used to provide cabins, and also the Nissen huts in which are the messrooms and the chapel; but until the accommodation block is built about 150 officers will continue to live at Keyham.

An important part of the training is centered on the work of the "Design and Make" teams—groups of officers who are required to design some piece of machinery such as a small pump and then, having designed it, to construct it and set it to work; an excellent all-round test of their success in absorbing both the theoretical and the practical sides of their education, and of their ability to think for themselves. In addition to the practical engineering

subjects and the development of craftsmanship, which together occupy a third of the time devoted to the complete course of instruction, every encouragement is given to those whose natural mechanical bent leans towards hobbies of a similar character. In one of the hangars, space is provided for the overhaul of cars—some of which, as acquired, appear to be much in need of it; though, at the same time, some supervision is exercised over the duration of the overhauls, for all officers are expected to take their part in the more usual recreations, for which facilities are provided on a generous scale. These include not only the customary games, but sailing—the College possesses two yachts, two whalers, and ten dinghies, with a boathouse on the Hamoaze—and also flying, which is practiced at the Flying Club at Roborough, where four aircraft are maintained for training purposes. Attached to the College there are also various motor gunboats, modern enough to be operational if the occasion should arise, which are available for practical instruction afloat.

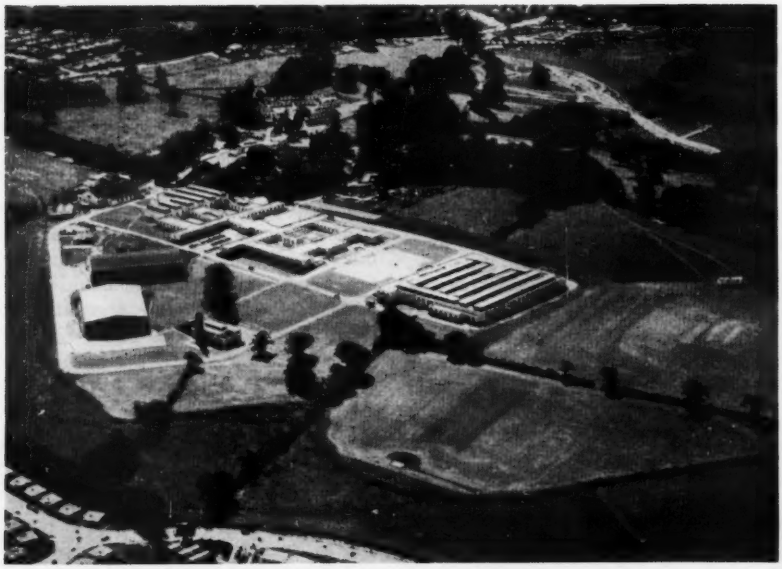
NEW SCHEME OF TRAINING

The scheme of training is designed to fit in with the new entry scheme which is to come into operation in May, 1955, under which all cadets will enter at about 18 years of age and will not go to sea in ships of the active Fleet until they have attained to the rank of Acting Sub-Lieutenant. Their training up to that stage will be centered on the Royal Naval College at Dartmouth, where the course will be divided into three parts. The first part, consisting of two terms, will provide initial instruction in the usual academic subjects and also in naval history and customs, divisional duties, elementary pilotage, boatwork, etc.

The second stage in the course will consist of one term to be spent at sea in a training squadron of frigates, minesweepers, and similar vessels, based on the River Dart. No formal instruction will be given during this term, the main purpose being to introduce the cadets to the conditions of life in seagoing ships; they will serve, therefore, as part of the rating complement of the vessels. In the final phase of the course, they will spend three to four terms on shore and in the training squadron under instruction in the principal professional subjects. Provision will be made, also, for experience to be gained in a naval air station.

At the outset of the new scheme, candidates will be entered, as at present, as Executive, Engineer, or Supply and Secretariat Cadets, and the training will be such that, when they first go to sea with the Fleet, they will be well grounded in the customs and traditions of the Navy and will know what is expected of an officer, and they will also have sufficient professional knowledge (in the official phrase) "to make them capable of efficient and intelligent understudy."

The young officer (E) who enters Manadon under either the existing arrangements or, in due course, under the new scheme, is destined to receive a training there which, in all essentials, is the equivalent of a university degree course except for the fact that no degree, as such, is conferred. The method is a combination of lectures and a university tutorial system, and lays upon the officer himself the responsibility of seeing that he profits to the utmost from the facilities and the instruction available. From the start, he is taught to think for himself. Certain periods are allocated to private study. Instruction in matters of discipline,



H.M.S. *Thunderer*, the Royal Naval Engineering College at Manadon, Plymouth, where officers are trained in marine, ordnance and aeronautical engineering.

and welfare and "officer training," are in the hands of the "term officers," a term being a group who have joined together as cadets and who remain together throughout their training. The staff of the College, both training and administrative, are nearly all serving officers from the Engineering and Instructor Branches, under the command of a Captain (E)—at the present time, Captain (E) C. Gatey, C.B.E.

Attendance at lectures and in the workshop courses is normally compulsory. The program also allows for laboratory and tutorial periods, the tutoring system having the dual purpose of affording individual help and guidance to the officers under instruction, as and when they may require it, and of enabling the staff to assess the progress and ability of the officers as they work through the course.

The general aim is to produce naval engineer officers who are efficient both as officers and as engineers, the minimum professional standard required at the end of the course being that of a university engineering degree. The final examinations are set and marked by external examiners from the Royal Naval College at Greenwich, and universities such as those of Cambridge and London. The equivalent course at a university would take three years, but the officers at Manadon have shorter leave periods than the vacations at a university and are kept working at high pressure; by this means it is possible to cover the required syllabus in two years. The final examination carries exemption from Sections A, B and C of the associate-membership examination of the Institution of Mechanical Engineers, and the officers who pass it are eligible for immediate election to the Institution as graduates.

SPECIALIZATION

Towards the end of the basic course of two years, the officer under instruction makes his choice between the three branches—marine engineering, ordnance engineering, and aeronautical engineering—and thenceforward his studies are directed to eventual specialization in the department of his choice. In theory, the selection is not absolutely unfettered, for consideration must be given to the numbers laid down by the Admiralty for the "establishment" of each Branch, but in practice nearly all the officers do enter the Branch of their first choice.

On passing out from the College, officers are appointed to seagoing ships of the Fleet, to obtain their watchkeeping certificates, and then they return to the College for their specialists' courses. Those who are specializing in marine or aeronautical engineering spend 12 months at sea, but those who are taking up ordnance engineering only eight months, as they subsequently divide a further eight months between H.M.S. *Excellent* and H.M.S. *Vernon,* the gunnery and torpedo schools, respectively, at Portsmouth.

Immediately after returning from sea to the College, all the officers, irrespective of their specialization, have to take what is termed a "consolidation course" of five weeks in marine engineering, ending with an examination, before starting on their specialist courses. It might be better described, perhaps, as an "acclimatization course," to re-introduce them to an atmosphere of study (and to show them that there is still something to learn) after their very different experience of life on board ship.

The specialist courses, which are at a post-graduate level, not only extend and apply the scientific and technical instruction previously given, but introduce new subjects such as aerodynamics and electronics. The main part of the course is designed to impart the ability to operate, maintain and overhaul the equipment used in marine, ordnance and aeronautical engineering, and, by constructive criticism and, on occasion, actual design proposals, to improve it. Concurrently, the administrative instruction also continues, for the higher posts in the Engineering Branch are open to all engineer officers, without regard to their specialization, and they must be trained accordingly.

On the conclusion of a specialist course, an officer is fully equipped to fill appointments in his particular Branch. A small number undergo further advanced courses to fit them for posts in design, research and development in Admiralty departments or elsewhere; these cover two years and are taken at the Royal Naval College, Greenwich, in the case of marine and ordnance engineering officers. Those who have specialized in aeronautical engineering take their advanced courses at the College of Aeronautics, Cranfield. If physically fit, they can also volunteer for flying duties, in which case, after qualifying as pilots, they will be employed for a time in flying Service aircraft, with the opportunity to rise to the command of a squadron or to become test pilots. Similarly, an officer who specializes in marine engineering can volunteer for service in submarines.

However officers may specialize at or after Manadon, they are all fully-trained naval officers and mechanical engineers, and, as stated above, are eligible for promotion to the higher posts in the Engineering Branch of the Navy. Most of these posts are concerned primarily and directly with marine engineering, and for this rea-

son both the ordnance and the aeronautical engineering officers will return to sea at various stages in their careers for general service so that they do not lose touch with marine engineering practice and developments.

OTHER COURSES

The estalishment at Manadon has other functions to perform, in addition to the training of engineer officers, who enter it as cadets or return to it from sea. It has also to provide technical training for Sub-Lieutenants (E), who are promoted from the lower deck, for the small number who enter directly from universities, for Constructor Sub-Lieutenants and Probationary Assistant Electrical Engineers, for what are now known as Branch Officers (formerly Warrant Officers), for National Service officers, and, finally, for the provision of courses in naval administration for officers who have attained the rank of Lieut.-Commander (E) and of various specialized refresher courses. Manadon also accepts engineer officers of Commonwealth navies and, if required, officers of friendly powers.

The Sub-Lieutenant (E) who is promoted from artificer is first given a cultural course of eight months at the Royal Naval College, Greenwich, and is then transferred to Manadon for a year, at the end of which he should attain the standard represented by the end of the basic engineering course. Thereafter his training is the same as that of the cadet-entry officers. Engineer officers who are taken in from the universities enter at Manadon and are given a course to familiarize them with naval life and to give them a basic knowledge of practical engineering before they are sent to sea or elsewhere for practical training. Then they, too, follow the program laid down for cadet-entry officers.

Branch officers undergo a professional course of six months at Manadon, which includes some introduction to the work of the specialized officers; and the National Service officers, who are promoted to Acting Sub-Lieutenant (E), R.N.V.R., after four months' service as ratings, do three months at Manadon before proceeding to sea to obtain their watch-keeping certificates or to a naval air station to obtain certificates of competency in aeronautical engineering. National Service officers are not trained in ordnance engineering.

It will be seen, therefore, that the whole conception of the establishment at Manadon is on a scale never before provided, and fully justifies the description of H.M.S. *Thunderer* as a naval engineering university.

LAMINAR HEAT TRANSFER IN RECTANGULAR TUBES WITH COMBINED FREE- AND FORCED-CONVECTION

L. S. HAN

THE AUTHOR

received his degree, Doctor of Philosophy, in June, 1954 from the Ohio State University where he has been, for the past five years, doing research work in fluid mechanics and heat transfer. He is currently a faculty member of the department of Engineering Mechanics and a research associate in mechanical engineering.

INTRODUCTION

Recently, there have been research activities indicating an increasing importance of heat transfer by combined free- and forced-convection especially in long narrow passages as encountered in various branches of engineering. Investigations have been undertaken particularly at NACA to study the combined effect of free convection and forced convection in such a configuration. Ostrach[1]* published a report of a theoretical investigation on the combined natural- and forced-convection heat transfer. The configuration treated in that report is a two-parallel-plate system and on account of the simplicity of the geometry, all possible variables can be included in the analysis and their physical significance explained. The method of analysis employed consists of a first approximation by dropping the aerodynamic heating term and then a second approximation by including the neglected heating term, the latter being computed from the first approximation and so on.

DEVELOPMENT AND SOLUTION

In this article we shall consider the flow in a long rectangular tube as shown schematically in Figure 1. A rectangular cross section is a more realistic configuration in actual application such as in a laminated heat exchanger.

( )*refer to similarly numbered references.

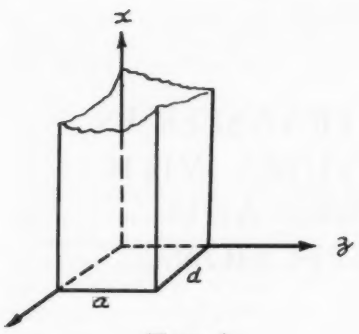


Figure 1

We shall retain all the assumptions made by Ostrach except that the aerodynamic heating term and internal heat generation are negligible. The main assumptions are grouped as follows:

(1). Constant physical properties.

(2). Laminar flow (fully established).

(3). Constant axial temperature gradient.

The solution indicated in this article will then be a first approximation and for more accurate value its successive approximations can be easily constructed by numerical methods.

Referring to Figure 1, the flow channel is defined by $y = 0,d$ and $z = 0,a$. We can, without going into details of the development of equation for which the reader is referred to reference (1), immediately write down the equations for one dimensional case (a two-parallel-plate system, for $a \to \infty$), neglecting heat generation and the aerodynamic heating, as

$$\frac{\partial^2 U}{\partial y^2} + \frac{\rho \beta f_x \theta}{\mu} = \frac{1}{\mu}\left(\frac{dP}{dx} + \rho_w f_x\right) \quad (1)$$

$$\frac{\partial^2 \theta}{\partial y^2} - \frac{\rho C_v A U}{K} = O \quad (2)$$

where,

A=temperature gradient, degree/axial length
C_v=specific heat
f_x=negative of body force in x-direction
k=thermal conductivity
P=pressure
T_o=wall temperature at x=0
T=fluid temperature
$T_{wo}=(T_o+Ax)$, wall temperature at x
U=velocity in x-direction
w=wall condition
$\theta=T-T_{wo}$
μ=viscosity
ρ=density
β=coefficient of expansion

We can rewrite equations (1) and (2) for a two dimensional case as,

$$\nabla^2 U = \frac{1}{\mu}\left(\frac{dP}{dx} + \rho_w f_x\right) - \frac{\rho \beta f_x}{\mu}\theta \quad (3)$$

$$\nabla^2 \theta = \frac{\rho C_v A U}{K} \quad (4)$$

where,

$$\nabla^2 = \frac{\partial^2}{\partial y^2} + \frac{\partial^2}{\partial z^2}$$

Equations (3) and (4) are the reduced hydrodynanmic and energy equations. The boundary conditions are as follows:

y=0,d;	z=0,a
U=0;	θ=0

It is realized that the thermal boundary condition so specified is an idealized one. There is some justification in some cases where highly conductive material is used to partition the flow channels. It might be further added that the condition of constant axial temperature gradient is quite closely approximated in counter-flow heat exchanges with nearly equal heat capacities for both fluid. This has been pointed by Eckert[2] and London and Kays.[3]

Returning to equations (3) and (4), one can combine these two equations into a single one as,

$$\nabla^4\theta=\frac{\rho C_v A}{k\mu}\left(\frac{dP}{dx}+\rho_w f_x\right)-\left[\frac{\rho^2 C_v \beta f_x A}{k\mu}\right]\theta \tag{5}$$

with the following as the boundary conditions:

$$\theta_w=0$$
$$(\nabla^2\theta)_w=0 \tag{6}$$

Note that the second condition stems from the condition that $U=0$ at the wall. The readers familiar with elastic theory of thin plates, the resemblance of equation (5) with the well-known Lagrange's equation is striking. This resemblance has been noted earlier by Marco and Han[4] who also suggested the use of analogous solutions to heat transfer problems.

Hence if one, using Fourier's expansion, writes the identity,[5]

$$\frac{\rho C_v A}{k\mu}\left(\frac{dP}{dx}+\rho_w f_x\right)=\left(\frac{16}{\pi^2}\right)\left(\frac{\rho C_v A}{k\mu}\right)\left(\frac{dP}{dx}+\rho_w f_x\right)\sum_{\substack{m=1,3,5\ldots\\ n=1,3,5\ldots}}^{\infty}\sum^{\infty}\frac{1}{mn}\sin\frac{m\pi y}{d}\sin\frac{n\pi z}{a} \tag{7}$$

and also assumes the solution θ as,

$$\theta=\sum_{m=1,3,5\ldots}^{\infty} a_{mn}\sin\frac{m\pi y}{d}\sin\frac{n\pi z}{a} \tag{8}$$

The undetermined coefficients a_{mn} can be readily established by substituting equations (7) and (8) into (5), then

$$a_{mn}=\frac{16PC_v Ad^4}{mn\pi^2 K\mu}\left(\frac{dP}{dx}+\rho_w f_x\right)\cdot\frac{1}{\pi^4(m^2+n^2\alpha^2)^2+\dfrac{\rho^2 C_v\beta f_x Ad^4}{k\mu}} \tag{9}$$

where,

$$\alpha=d/a$$

and consequently, the velocity U is given by equation (4) as,

$$U=\frac{K}{\rho C_v A}\nabla^2\theta$$
$$=\frac{16}{\mu}\left(\frac{dP}{dx}+\rho_w f_x\right)d^2\sum_{\substack{m=1,3,5\ldots\\ n=1,3,5\ldots}}^{\infty}\sum^{\infty}\frac{(m^2+n^2d^2)\sin\dfrac{m\pi y}{d}\sin\dfrac{n\pi z}{a}}{mn\pi^4\left[(m^2+n^2\alpha^2)^2+\dfrac{\rho^2 C_v\beta f_x Ad^4}{k\mu}\right]} \tag{10}$$

Note that Ostrach defines,

$$\frac{\rho^2 C_v\beta f_x Ad^4}{k\mu}=\left(\frac{1}{\alpha}P_r G_r\right)=R_A \text{ (Rayleigh number)}$$

where,

$\alpha = \frac{C_p}{C_v}$; $P_r = \frac{C_p \mu}{k}$ = Prandtl's number

Equations (9) and (10) can be used to define the Nusselt number from which the heat transfer coefficient can be calculated. Thus,

$$N_u = \frac{hD_e}{k}$$

where,

D_e = the hydraulic diameter = $\frac{2ad}{(a+d)}$

h = the heat transfer coefficient, defined by

$$h = \frac{q}{2(a+d)(t_w - t_m)}$$

t_m = the mixed mean temperature, defined by

$$t_w - t_m = \frac{\int\int U\theta dydz}{\int\int U dydz}$$

q = heat given off or received by the surfaces, defined by

$$q = \rho C_v A \int\int U A dydz$$

Hence,

$$Nu = \frac{ad}{(a+d)^2} \cdot \frac{\rho C_v}{k} \cdot A \cdot [\int\int U dydz]^2 / [\int\int U\theta dydz] \qquad (11)$$

Upon introducing equations (8), (9) and (10)

$$Nu = \frac{\frac{64}{(1+\alpha)^2\pi^2} \sum_{\substack{m=1,3,5\ldots \\ n=1,3,5\ldots}}^{\infty}\sum^{\infty} \left[\frac{m^2+n^2\alpha^2}{m^2n^2[\pi^4(m^2+n^2\alpha^2)^2+R_A]}\right]^2}{\left[\sum_{\substack{m=1,3,5\ldots \\ n=1,3,5\ldots}}^{\infty}\sum^{\infty} \frac{m^2+n^2\alpha^2}{m^2n^2[\pi^4(m^2+n^2\alpha^2)^2+R_A]^2}\right]} \qquad (12)$$

Note that equation (12) reduces to that given in reference (4) when $R_A = 0$. From equations (8), (10) and (12) for θ, U and N_u it becomes evident that for positive values of R_A, the flow field is stable. For negative values of R_A, the flow will not be stable when,

$$|R_A| > (1+\alpha^2)^2\pi^4$$

This, however, is only qualitative since the real process is much more complex than is pictured in the present paper.

It is interesting to note that when R_A is extremely large, an assymptotic solution for N_u exists, i.e.,

$$Nu = 8\left(1 - \frac{2\alpha}{(1+\alpha)^2}\right) \qquad (13)$$
$$R_A \to \infty$$

CONCLUSIONS

Consideration of two dimensional flow in rectangular cross sections with combined free- and forced-convection leads to the expression for the Nusselt number (equation (12)) which is essentially similar to that derived in reference (4) for a pure forced-convection flow. In the latter case, the Rayleigh's number is zero, and two formulas are identical to one another. The limiting values of the Nusselt numbers for large R_A are given by equation (13). The velocity and temperature fields are given by equations (8), (9) and (10) from which their point values and their derivatives can be computed from necessary numerical work. It is to be noted that the solutions need not be limited to rectangular cross sections. One can use the method presented

here and in reference (4) to the cases where the cross sections are non-rectangular. Two other cross sections can be easily and suitably treated, i.e., isoceles right-angled triangular, and semi-circular which occur frequently in heat exchanger construction.

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Official U. S. Navy Photo Released by Department of Defense, Washington, D.C.

USS *Nautilus* (SS-571), The Navy's First Atomic-Powered Submarine, on its initial sea trials.

SHIPBOARD MEASUREMENT OF INSULATION RESISTANCE OF ENERGIZED D.C. CIRCUITS

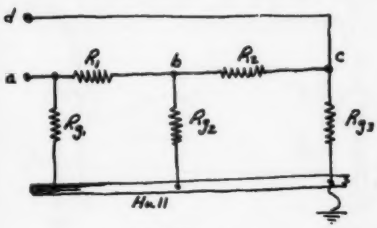
LIEUTENANT CARVEL HALL BLAIR, U. S. NAVY

THE AUTHOR

graduated from the U. S. Naval Academy in 1944. After graduation he served on USS *Wilkes-Barre* (CL 103) until his assignment to the Submarine School, New London, Connecticut in 1946. He then served on USS *Odax* (SS 484) as Electrical Officer during her conversion to the first Guppy (high submerged speed) submarine 1946-9 and on USS *Grampus* (SS 523) 1949-50. Then attended the Naval Postgraduate School. Presently is Executive Officer of the submarine USS *Wahoo* (SS 565).

Any discussion of insulation resistance must begin with definitions of the terms *ground* and *insulation resistance*. As used in this paper, they have the following meanings:

A *ground* is a path capable of carrying current between the hull of a ship and an installed electric circuit. In Figure 1, R_{g1}, R_{g2}, and R_{g3} are grounds.



Figures 1

Insulation resistance (R_g) of a circuit is the equivalent ohmic resistance of all parallel paths connecting the circuit to ground. For circuit *abcd* in Figure 1, the insulation resistance is found by the familiar formula:

$$\frac{1}{R_g} = \frac{1}{R_{g1}} + \frac{1}{R_{g2}} + \frac{1}{R_{g3}}$$

Lest these definitions be considered trivial, one must realize that it is impossible to measure the precise value of R_g for the circuit shown. It is possible to measure a resistance whose value differs from R_g only by terms involving R_1 and R_2. It becomes necessary to know the conditions under which this difference is negligibly small. To do so, one must consider the measuring circuits available.

Most instruments which can be used to measure R_g of deenergized circuits cannot be used to measure R_g of energized circuits. Meggers, ohmeters, and Wheatstone bridges, for example, either yield incorrect measurement or may be damaged if connected between ground and a live circuit. There are three general schemes for determining R_g of energized circuits. They involve, respectively, the incandescent lamp, the voltmeter, and the balanced ohmeter.

The simplest circuit consists merely of a lamp which is connected alternately from ground to the positive and negative terminals of the circuit. In Figure 2, with the switch in the (+) position, current flows from the (+) terminal through the lamp to ground, through ground to R_g, thence through R_1 back to the (+) terminal. The bulb lights, its brilliance depending on the values of E, R_g, and R_1. With the switch in the (−) position, the bulb again lights, but now its brilliance depends on the values of E, R_g, and R_2. Thus the lighting of the "ground detector" bulb is qualitative proof of the existence of a ground: the brighter the lamp, the lower the ground resistance. It is impossible accurately to determine either the location or the value of the ground;

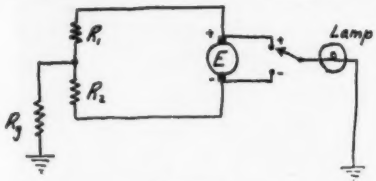


Figure 2

the presence of several grounds or of a ground inside the voltage supply can give misleading indications.

A more elaborate device consists of a d'Arsonval type voltmeter, again connected alternately from ground to the positive and negative terminals of the power supply. (See Figure 3.) It can be shown that the value

$$R'_g = R_m \left[\frac{E}{V_p + V_n} - 1 \right] , \text{ where}$$

R_m = resistance of voltmeter,
E = battery terminal voltage,
V_p = absolute value of voltmeter reading with switch in (+) position, and
V_n = absolute value of voltmeter reading with switch in (−) position, is very close to the true value of R_g. In fact,

$$R'_g = R_g + \frac{R_1 R_2}{R_1 + R_2} .$$

For the majority of shipboard power circuits, the second term on the right is negligible and we can legitimately make the approximation that

$$R_g \approx R_m \left[\frac{E}{V_p + V_n} - 1 \right]$$

If there are more than two grounds, especially in the power supply, this proof becomes exceedingly tedious. It should be intuitively evident, however, that if other resistances are low, including those of the voltage source, the above formula for R_g will give an engineering approximation of the true insulation resistance. Such is usually the case in power circuits, but not necessarily in communication circuits.

Errors are often made by operating personnel in using the ground detector voltmeter. Perhaps the most common, and one of the most dangerous, is to use an incorrect value of R_m. In some installations it is possible to reduce R_m from 50,000 ohms to 5,500 ohms by depressing a button. If the reduced values of V_p and V_n which inevitably result are used in conjunction with the higher value of R_m, an incorrect value of R_g will be computed. The error will

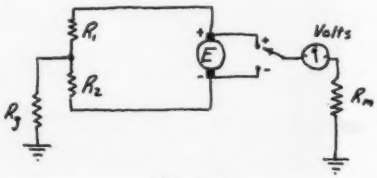
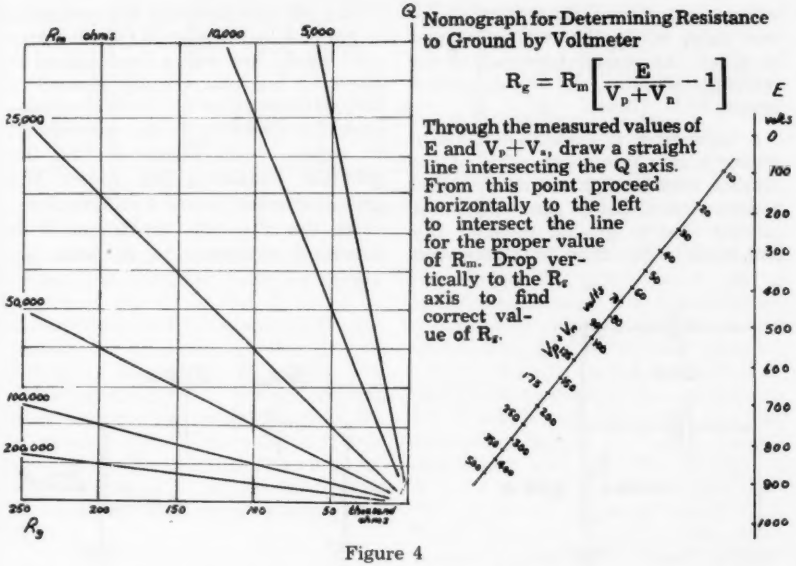


Figure 3



Nomograph for Determining Resistance to Ground by Voltmeter

$$R_g = R_m\left[\frac{E}{V_p+V_n} - 1\right]$$

Through the measured values of E and V_p+V_n, draw a straight line intersecting the Q axis. From this point proceed horizontally to the left to intersect the line for the proper value of R_m. Drop vertically to the R_g axis to find correct value of R_g.

Figure 4

be in the dangerous (i.e. high) direction. The use of the nomograph in Figure 4 will help avoid such errors.

A second error occurs when several batteries can be shifted from parallel to series. Figure 5 (a) shows the correct connection for two batteries in parallel; Figure 5 (b), for two batteries in series. The connection shown in Figure 5 (c), which exists in some instal-

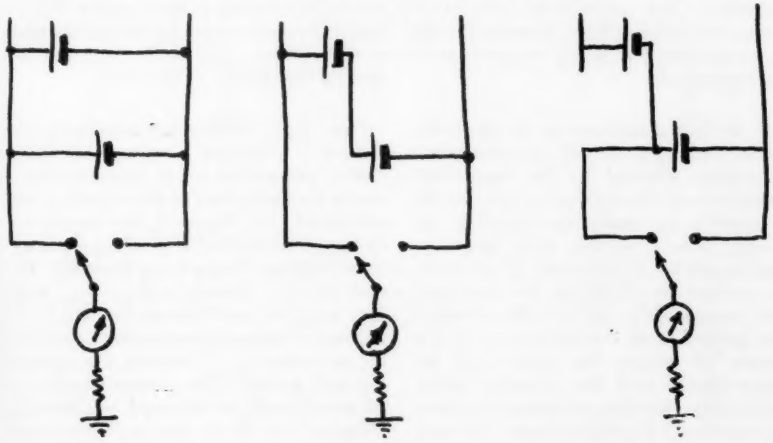
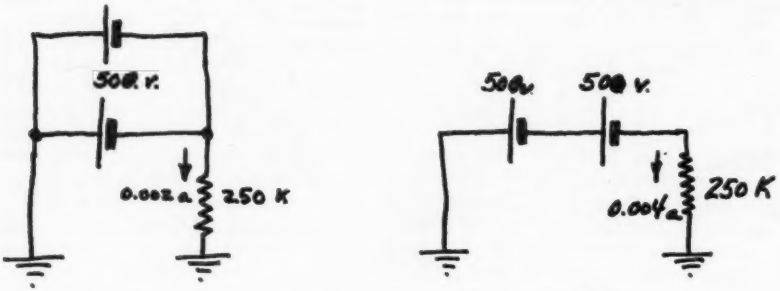


Figure 5

(a) correct (b) correct (c) incorrect

lations, is incorrect. To determine the true value of R_g, the voltmeter must be across the outside terminals of the series combination, not across interior points.

A third error is sometimes made in setting a safe minimum value of insulation resistance in a series-parallel battery circuit. Let us assume that the current flow to ground through any path must be less than 2 milliamperes. If two 500 volt batteries are connected in parallel, as in Figure 6 (a), this current would flow with a dead ground at the (+) bus and a single ground of 250,000 ohms at the (−) bus. Now suppose that the batteries are reconnected in series, as in Figure 6 (b), the grounds remaining the same. The ground current is now 4 milliamperes, twice the allowable maximum. It is therefore necessary to increase the



(a) (b)

Figure 6

minimum allowable resistance to ground to 500,000 ohms. Thus a safe value of resistance for a parallel connection is not necessarily safe for a series connection. The value set should be reexamined if series connection is contemplated.

A serious disadvantage of the voltmeter method is that R_g, as measured, is always affected by the insulation resistance of the voltage source, which is usually an unknown quantity. In many cases it is this very quantity that must be determined. If a motor is running on a battery, for example, it is impossible to tell directly whether the ground is in the battery or in the motor. Of course the motor may be disconnected and the circuits tested separately, but this procedure requires an additional insulation tester. Several forms of ohmeter have been developed to overcome this difficulty. They can be used on energized or deenergized circuits and have the added advantage of reading directly in ohms. They all employ a separate power supply whose insulation resistance can be maintained at a very high value and which can be readily checked.

One type, shown schematically in Figure 7, employs a voltage divider. Here a potentiometer is connected between the polarities of the circuit to be measured. (In Figure 7, the circuit to be tested consists of a running generator of voltage E supplying loads R_1+R_2 and R_3+R_4. Grounds R_{g1}, R_{g2}, and R_{g3} exist at the points shown.) A milliammeter and protective resistance R_m are connected between the center tap and ground. The potentiometer is adjusted until a balanced position is obtained in which the milliammeter reads zero. A known voltage V (with infinite resistance to ground) is then

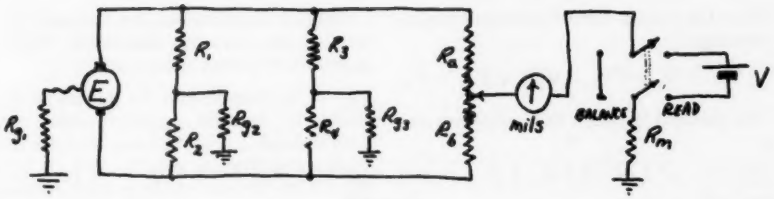


Figure 7

connected in series with the milliammeter, and a current flows. By proper choice of $R_a + R_b$ and R_m, and by calibration of the milliammeter, the deflection can be made to indicate ohms resistance to ground. This value will differ negligibly, in power circuits, from the true insulation resistance R_g. A proof, by the Superposition Theorem, will show that this difference involves terms in R_1, R_2, R_3, R_4, R_a, and R_b which are small in comparison with R_g.

A second type of ohmeter utilizes a constant voltage power pack with a variable voltage tap. Figure 8 illustrates this circuit schematically. The circuit consists of the power pack plus a switch connecting it to a milliammeter, a protective resistor, and the circuit under test. In the "Adjust" position, (Figure 8 a) the tap is set so that no current flows through the milliammeter. Kirchoff's voltage law shows that the voltage tapped off, V_{ba}, is of equal magnitude and opposite polarity to V_{de}, the voltage of the energized portion of the circuit being tested. Thus $V_{ba} = -V_{de}$, or $V_{ab} = V_{de}$. The switch is then thrown to the "Read" position (Figure 8 b). The e.m.f. impressed on circuit abcdef is then $V_{bc} + V_{de}$. But $V_{bc} = V_{ac} - V_{ab}$.

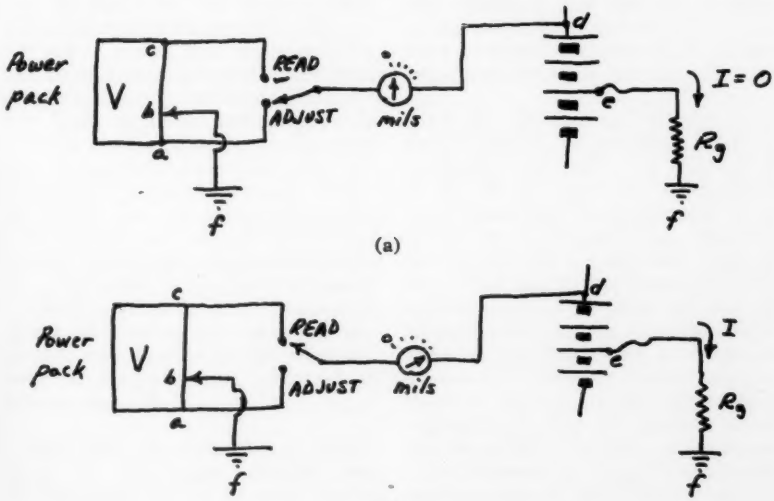


Figure 8

Then the circuit e.m.f. becomes by substitution,

$$V_{bc} + V_{de} = (V_{ac} - V_{ab}) + V_{ab} = V_{ac}$$

The current through the milliammeter is

$$I = \frac{V_{ac}}{R_p + R_g}$$

With V_{ac} and R_p known constants, R_g is obviously an implicit function of I. The meter can thus be calibrated in ohms R_g, provided again that load and battery resistances are small compared to ground resistance.

Table I summarizes the characteristics of the circuits discussed. Three significant points stand out:

(1) It is impossible to measure R_g precisely, but an approximation can be obtained. In most power circuits the error is negligibly small.

(2) Direct ohmic readings and greater accuracy are obtained at the expense of greater cost, maintenance, and complexity.

(3) Familiarity with the measuring instrument and the circuit measured is essential to avoid dangerous errors.

TABLE I *Summary of circuit characteristics*

	Lamp	Voltmeter	Balanced Ohmeter
Cost	Little	More	Most
Type of reading	Qualitative	Quantitative	Quantitative
Portable?	Yes	Yes	No
Reading direct in ohms?	No	No	Yes
Applicable to deenergized circuit	No	No	Yes
Maintenance	Least	Little	Moderate

SOME ELECTRONIC INSTRUMENTS FOR MARINE TURBINE RESEARCH

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INTRODUCTION

Engineering research and development demand accurate and convenient instrumentation. Many measurements may be made by well-known, well-tried methods which have been known for many years, but more recently electronic instruments have been used to measure many mechanical quantities. Some engineers lack enthusiasm for this development on the grounds that the instruments are probably unreliable and certainly incomprehensible. Others use electronic instruments, presumably in an attempt to be fashionable, where the measurement might be done better by orthodox methods. Properly applied, however, electronic circuit techniques can make an important contribution to the wide variety of measurements required in turbine research.

Soon after the formation of Pametrada an Electronics Section was started to design and construct measuring instruments. Initially work was on apparatus for noise and vibration measurement, for which electronic instruments are particularly appropriate. However, other branches of turbine research can usefully employ similar methods and a wide field has been covered. No electronic control apparatus has been considered.

Almost all the instruments have been designed for operation of non-specialists and deliver a reading in lb.-ft., °F., in. per sec. or whatever mechanical unit is appropriate. Even where the instruments are used by specialists with a good knowledge of the circuit details this design principle is important; it obviates the risk of transcription errors, is less fatiguing to use and, most important, it enables an answer to be obtained immediately instead of waiting for analysis of photographic or other records. For vibration diagnosis at sea this is of particular value because each stage in the investigation may be based on the results already obtained and supplementary measurements made as required. Thus

there is no possibility of discovering, after the analysis of records ashore, that important data are lacking because their relevance was not appreciated at the time. Instruments for analysis of mechanical quantities are usually easier to devise when the mechanical quantity is translated into electrical terms. Mathematical operations such as addition, integration and differentiation may easily be performed on electrical functions of time, a facility which is exploited in many of the instruments described.

Work has been restricted to the development of instruments which are not commercially available: those described in the paper are not (or at the time were not) obtainable "off the shelf." Some of them were quite original and have recently been produced commercially because they are of wide application, while others were more specialized and developed to provide greater accuracy than is normally obtainable. All equipment is made on the Station and designed accordingly. It is unusual for accurate mechanical work to be necessary, and a factor which sometimes makes electronic methods preferable to alternatives is that the instruments are assembled from stock components which are in cheap and plentiful supply.

There are several reasons for using electronic instruments in marine turbine research. Sometimes they provide the only means of making a measurement: the instruments described for noise and vibration analysis are in this class. The fact that they have no moving parts implies that operations may be performed at very high speed, which means, amongst other things, that high accuracy may be achieved in rapid measurements: the wave correlator and r.p.m. indicator depend on this. A prime advantage is convenience. First, it is often possible to connect instrument to machine without any mechanical connection to the moving parts. The transducer—the device which produces an electrical output which is a function of the mechanical quantity to be measured—may be small and easily mounted on (or near to) the part of the machinery under investigation. The small output obtained may then be made sufficiently large for indication or analysis by valve amplifiers. Interconnections are easily installed flexible cables. Second, a group of instruments may be interconnected in a variety of ways as the situation demands.

When instruments have been developed with the high accuracies implied above, arrangements must be made for their calibration. Both microphones and vibration pick-ups require initial calibration and frequent checks, and the echo-free room designed for microphone calibration is described in some detail. A Standards Room houses the means for calibrating thermometers, various sub-standard instruments and a standard frequency source.

The paper gives an account of some electronic instruments that have been developed in Pametrada Research Station. For reasons of space details of their use cannot be considered. Some fairly orthodox instruments developed for special purposes have been omitted so that more interesting ones may be considered in greater detail. Brief mention is made of developments now in progress. Since the majority of electronic instruments are used for noise and vibration research this subject will be considered first.

INSTRUMENTS FOR NOISE AND VIBRATION MEASUREMENT

Estimates of noise and vibration intensity are difficult to make without the help of instruments. The judgment of any observer varies from day to day and two observers frequently disagree about the relative degrees of roughness of two machines, or of the same machine under different conditions. Electronic instruments are well suited to making measurements of this kind because convenient pointer readings of the relevant quantities may easily be obtained. In addition to the measuring instruments, amplifiers are usually required to provide a larger signal than the pick-up or microphone generates, to alter the impedance level, or to integrate the output of accelerometers to give a signal proportional to vibration velocity. However, a great deal may be done with simple self-contained instruments and one of the first tasks of the Section was to construct a portable vibration meter. This meter measures total vibration and is small enough to carry easily. The circuit is a straightforward 3-valve amplifier using fairly large batteries to give long life. It has an integrator between the first two valves. The scale reads $0-5\times10^{-3}$ in. per sec. vibration velocity and a stepped attenuator caters for vibration velocities much higher than those met in practice. A Rothermel VP5 vibration pick-up is used. This simple instrument has been much used and found satisfactory. Recently some work has been done to develop a cheap simple instrument covering the 40-decibel range of noise levels above 80 decibels—the "noisy machinery" region. It is to have no controls other than an "on-off" switch and should be small enough to go in the trouser pocket. A vibration pick-up could replace the microphone when vibration measurements are required. Preferably, the meter will use only one valve, or if possible, none at all. One or two instruments have been built though an entirely satisfactory model has not yet been produced.

Comprehensive measurements demand more elaborate equipment, and pre-amplifiers are nearly always necessary between transducer and measuring instrument. The transducer is usually remote from the measuring equipment; it often delivers a small signal at a high impedance level. The pre-amplifier increases the signal level and also acts as a transformer between the high pick-up impedance and the low cable impedance. When using accelerometers an integrator may be switched in to give an output signal proportional to vibration velocity. Various amplifiers have been used in the past but two main types are now available.

The mains-driven amplifiers have a voltage gain of 40 db. $\pm$0.1 db. with variations of loop gain of $\pm$20 per cent. Panel switches give a choice of low or high input impedance, optional high-frequency cut-off at about 10 kc/s and integration with a choice of frequency below which linearity falls off. A stepped attenuator is not provided in the amplifier case. So that battery supplies may be used, standard mains valves with low consumption heaters are used, though careful selection of the valves for the first two stages is necessary to achieve a low enough hum level.

Miniature battery amplifiers are used with high-impedence accelerometers, which do not allow the use of more than a few inches of connecting cable. Self-contained cathode-followers have been constructed with an input impedance of 100 megohms to deliver output at low impedance to small 6 db. battery amplifiers. These have good gain stability and low noise and are suitable for feeding a long cable.

Two such amplifiers, with all batteries and a spare valve, are built into each box. If it is necessary to integrate the signal one of the mains-driven amplifiers is used. In the near future the accelerometer heads will be modified to house a cathode follower so that the short connecting lead may be eliminated and maximum transducer sensitivity achieved. Since the users sometimes wish to work with pick-up outputs of about 10 microvolts, all opportunities of obtaining more output must be taken.

Brief mention may be made here of a large amplifier built to drive a vibration motor. It is sometimes necessary to vibrate a structure artificially and examine its behavior. Investigations have been made, for example, into the nodal patterns obtained on large gear wheels. One way of doing this is to use a moving-coil vibration motor supplied from a power amplifier and variable-frequency oscillator. An amplifier with an output of 1 kVA. has been built complete on a trolley with its driver amplifier, impedance matching unit and power supplies. The circuits are orthodox, using class AB_2 triodes in the output stage.

WAVE ANALYZERS

One of the first electronic instruments required was a good wave analyzer to analyze complex vibration and noise wave-forms. Two main types are available; heterodyne and degenerative analyzers. The former has a number of disadvantages. From the Association's point of view the most important is that the presence of a component frequency is indicated only by a meter reading; the filtered component is not available at the output terminals of the instrument. For this reason the filtered component cannot be related by ear to the known characteristics of the machinery, and there is no possibility of subjecting it to further treatment in instruments for automatic analysis (one of which will be described in the next section).

Degenerative analyzers are feedback amplifiers with a tunable resistance-capacitance filter in the feedback path. A known circuit [1] was a parallel-T network in the negative feed-back path. The series impedance of the network, and therefore the gain of the amplifier, is a maximum at tune frequency. One disadvantage of the circuit is that its response does not fall away at a constant rate from resonance—the resonance curve has wide skirts, and the selectivity factor, Q, becomes poor at frequencies not far from resonance. Improvements were considered, and a new 8-element selective network was devised which, it may easily be shown, gives an analyzer of constant Q[2].

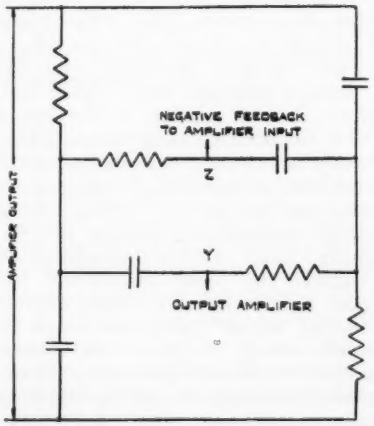

Fig. 1—Wave analyzer frequency-selective network.

[1] See bibliography.

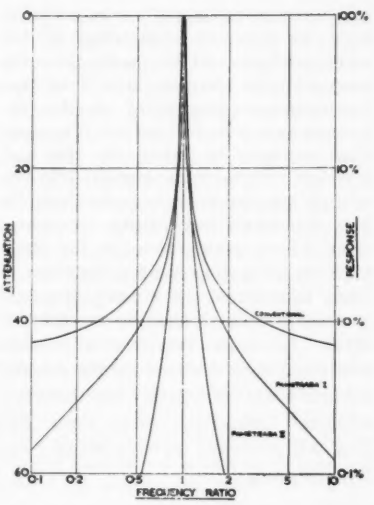

Fig. 2—Wave analyzer selectivity curves.

An instrument was designed to use this new circuit which is shown in Fig. 1. Its tuning range is from 30 c/s to 10 kc/s and five values of Q are provided, ranging from 100 to zero, in which condition the analyzer becomes a straightforward amplifier with a meter to read the average value of the input. The selectivity curve of the instrument is compared with that of a conventional analyzer using a parallel-T circuit (Fig. 2). Great use was made of the analyzer, which gave better performance than commercially available instruments. Experience showed a few desirable improvements: measurements on machinery never deal with steady quantities and highly selective analyzers become difficult to use when the input frequency wanders about the mean. If selectivity is reduced, discrimination against unwanted components becomes poor. The solution, of course, is to provide a selectivity characteristic with a flat top and steep sides so that small variations in input frequency are accommodated and adequate rejection of unwanted components maintained. Another desirable improvement, as always, was a decrease in size and weight. Work was put in hand on both problems. The former and more important, has some bearing on work on a low-frequency analyzer, and will be considered first.

One method of obtaining a band-pass amplifier (that is, one whose response curve has a flat top and steep sides) is to use stagger-tuned circuits. It is clear that if the output of one analyzer is fed into another, tuned to a slightly different frequency, the combined response curve will have a fairly flat top (possibly with slight humps) and steep flanks. An experimental band-pass analyzer with a small frequency range was built and tested. This was used as the basis of a design for a complete analyzer with wide frequency range and variable selectivity. An analyzer of this kind has advantages at low frequencies. With the original single-circuit analyzer it may be shown that the build-up time is roughly Q cycles. For example, if a signal at 10 c/s is applied to an analyzer with $Q=50$, steady response is obtained in 5 sec. This time may be considerably reduced by using two analyzers in cascade. Suppose that a cascade analyzer is made from two similar analyzers each with $Q=Q_2$ and is so designed that its response at half tune frequency is the same as that of a single analyzer with $Q=Q_1$. Then it may be shown that the build-up time of the cascaded amplifier is only $\sqrt{Q_1}$, and also that $Q_2=0.8\sqrt{Q_1}$. In the example quoted above the build-up time would be reduced to less than one second.

Work on wave analyzer II (with two analyzers in cascade) was carried on, though pressure of other work prevented progress with the low-frequency instrument. The cascaded analyzers

are tuned by Wien bridges, instead of the 8-element networks mentioned above, to reduce the number of variable resistors required for tuning. Development at Pametrada was dropped when the design was adopted as the basis for a commercial instrument, which was first used by the Association in January, 1949.

On recent trials magnetic tape records of noise or vibration have been made. A method of analyzing these records automatically is being considered, the analysis being written on a standard pen-recorder. A representative sample of magnetic tape is made into a loop and played back at gliding speed through a wave analyzer which is tuned to a fixed frequency. As the playback speed varies, each recorded frequency component is converted, in turn, to the tune frequency of the wave analyzer. At the same time the pen-recorder chart is moved so that the corresponding point on its frequency axis is under the pen. The analyzer output is taken to the pen through a logarithmic amplifier to obtain an amplitude scale in decibels. To give the usual logarithmic frequency scale it is necessary to make the playback speed a logarithmic function of time, which may be closely approximated by a fairly simple mechanical drive. Analysis requires a certain minimum time dictated by the desired accuracy and discrimination between adjacent frequency components; this time will probably be between one and two minutes.

WAVE CORRELATORS

The wave analyzer described in the previous section enables a complex vibration wave-form to be separated into its components. If, however, the analysis is to be more than an interesting academic exercise it is necessary to identify the source of each component. In general, the principal components are related in frequency to the speed of one of the shafts in the machinery and may be due to tooth contact in gears, misalignment of couplings, bent shafts, etc. In simple cases the source of vibration may be identified from the analyzer frequency dial and shaft speeds of the machinery, by looking for significant relationships between them. Unfortunately, instrument scales are of limited accuracy and engine r.p.m. unsteady, which leads to uncertainty whether a given frequency is an exact (and identifiable) multiple of a given shaft speed. Moreover, the search for correlation takes some time. A powerful means of identification in geared machinery is provided by the Pametrada wave correlator. This instrument indicates at once whether the filtered component (called the "signal") is an exact integral multiple of the speed of any of the shafts connected (electrically) with it. With a little elaboration it will indicate the order of the multiple, and thus give a complete analysis of the vibration wave-form that may be written down at sight.

The principle of operation was suggested by consideration of a type of Fourier integrator in which the filtered wave from the analyzer would be multiplied by a narrow square pulse of shaft frequency containing equal harmonic components up to about the 600th. The practical difficulties were discouraging, and it was seen that correlation could be indicated in a rather different way. The signal is examined once per revolution of the shaft at an instant defined by a pulse from a pick-up on that shaft. The polarity of the signal is assessed; if the signal frequency is an integral multiple of shaft

speed, the instrument will always "see" a half cycle of the same polarity. (The amplitude of the signal at the instant of examination, which sometimes shows fairly large random variations, is of secondary importance.) Two indicator lamps are provided, one of which flashes on positive, and the other on negative half cycles. Therefore if the signal and shaft frequencies are correlated only one lamp will flash; which lamp is immaterial. At any other signal frequency the lamps will flash in more or less random order, showing no correlation. At sufficiently high speeds correlation results in apparently continuous lighting of one lamp, and absence of correlation causes simultaneous lighting of both. Several shafts may be monitored at the same time by providing additional pairs of lamps and their associated circuits. Wave correlator I has 4 channels suitable for simultaneous examination of the low, intermediate and two high-speed shafts of a 2-cylinder double-reduction, geared turbine installation.

The next step, after the all-important one of detecting correlation, is to determine the order of the multiple. This may be done by displaying the signal on a cathode-ray oscillograph whose time base runs at shaft speed. The number of cycles on the screen is the order of the multiple. Where the number is not too large, it may be counted. Usually the amplitude is unsteady, which makes counting difficult; in any event the small screen of the cathode-ray oscillograph will not resolve the individual cycles of a high multiple. The highest correctly counted by the Authors on a sea trial was 40; usually the highest possible figure is about 20. To deal with high multiples an electronic counter was constructed to count the number of cycles of the signal which occurred between two successive shaft pulses. Spot readings were taken when required by pressing a button. A more detailed discussion of the use of the wave correlator is given elsewhere.[3]

The operation of wave correlator I may be followed with reference to a block diagram, Fig. 3. The signal is amplified and then phase-split so that two wave-trains are obtained 180° out

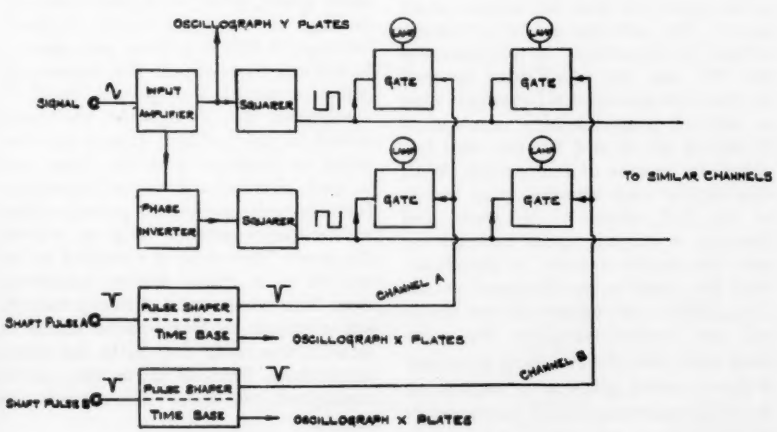

Fig. 3—Wave correlator: circuit arrangement.

of phase. Each is rectified, which produces a roughly square wave in each instance, negative-going from earth. The shaft pulse triggers a thyratron time-base generator to provide the c.r.o. trace mentioned above, and a short negative-going pulse is obtained once per revolution from the time-base flyback. The coincidence detectors, or "gates," are double triodes which provide a positive output pulse when both control grids are simultaneously negative. These circuits examine the signal for its sign, and one or other of the gates gives a positive output pulse, depending on whether the examining pulse arrives during a negative or positive half-cycle. The positive output pulses trigger neon flash tubes associated with each gate. The neon tube grids lose control after firing; the anode voltage, obtained from an RC circuit, gives a short flash and has a time constant such that the anode voltages rise to the necessary value before the next triggering pulse occurs. It will be seen that values of R and C identical to those in the thyratron time bases may be used, and the two sets of resistors and capacitors (per channel) are switched by means of the same control to cater for various shaft speeds. The switches are by no means critical in adjustment. In addition, the two RC sets are subdivided so that the time-constants of all channels may be altered proportionally in steps of 20, 30, 45, 60, 80 and 100 per cent full speed, by means of one switch. With this facility each channel may be set to the full speed of its shaft and changes of engine speed catered for with one master control. A switch allows the signal to be displayed on the c.r.o. with a time base from any channel, and another displays the pulse from each shaft for checking purposes. A check switch gives an overall check of circuit operation and in particular of the flash tubes, which have been found to have a short life. The other knobs are gain controls for the signal and pulse amplifiers.

After minor modifications the correlator has given invaluable and reliable service during several years of constant use, and experience suggests no change in the principle of operation. A similar instrument with 5 channels has been built with some minor changes to simplify construction and maintenance. In an attempt to improve the indication another 5-channel correlator with many circuit changes was built. In particular it was desired to replace the flashing lights by some type that would give a steady indication. This could be achieved by arranging that the anode time constant in the original instrument was such that the lamp would be extinguished just before the next triggering pulse was due; that is, of course, not practical. In the new instrument a neon lamp was chosen which lights brightly with a small steady current, and the circuit is arranged so that when either lamp in a pair is fired it extinguishes the other (this is an orthodox gas-triode scale-of-two). Thus correlation results in the steady lighting of one lamp instead of flashing at shaft speed, which is a particular advantage on low-speed shafts. A disadvantage is that one lamp per channel is always lit, even in the absence of pulse or signal, though this could be eradicated by a suitable electronic switch in the h.t. line. It was also decided to dispense with the time base in each channel and to use a straightforward pulse generator giving a pulse a few micro-seconds long to operate the gates. This pulse is switched as required to a single Miller integrator time base whose rate of sweep may be pre-selected by front panel knobs in exactly the same way as in the other correlators. Thus so far as the user is concerned, the three instruments are similar.

Since the primary function of a wave

correlator is to give a yes-or-no indication of correlation between a signal and a shaft speed, it is clear that a simple instrument may dispense altogether with a time base. A single-channel version of the last mentioned instrument has been built, designed so that it may conveniently be combined with others to examine multi-shaft machines. Each unit has its own power supply and both signal and standardized pulse are available on P.O. jacks so that they may be applied to a counter or cathode-ray oscillograph.

ROTOR BALANCING INSTRUMENT

A method of measuring turbine rotor out-of-balance forces is being developed with the ultimate aim of carrying out the measurement with the turbine mounted in its own bearings on board ship. The instrument proposed, shown in Fig. 4, would enable balancing to be carried out at a rotor speed as low as 500 r.p.m. and would indicate the phase-angle of the force with respect to a known fixed point, and also the amplitude of the force. As is common to other vibration measuring apparatus the instrument would have two principal parts; the pick-up and the instrument.

The vibration pick-up will take the form of a mass-spring system whose frequency would be tuned to 8.3 c/s (500 r.p.m.) and this operating into a velocity type of electro-mechanical pick-up would be mounted on one of the turbine bearing pedestals. This arrangement has several desirable features, being tuned to the balancing speed, vibration frequencies due to oil criticals existing on the pedestal are not transmitted to the instrument, and the sensitivity of the pick-up is increased by the Q of the system. In some systems of balancing it is usual to use the mass of the electrical pick-up as part of the mass-spring tuned system. Such a method, however, produces

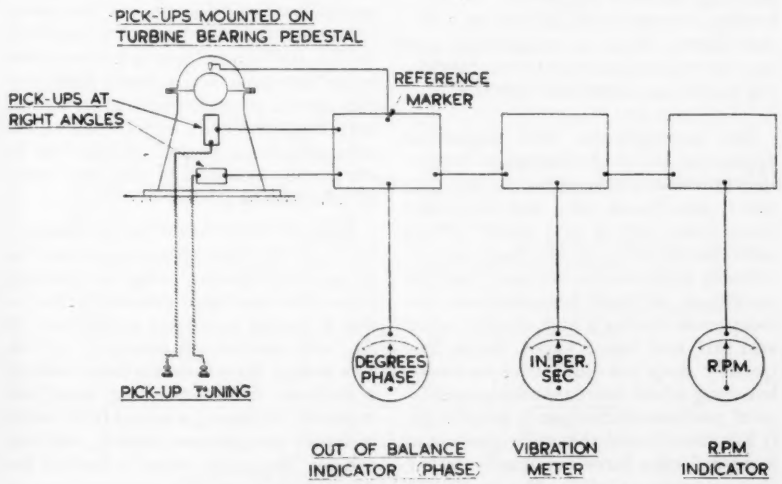

Fig. 4—Proposed out-of-balance indicator.

large errors of phase-angle which are not desirable in this application.

The voltage generated by the electrical pick-up will be applied to an electronic phasemeter which will indicate the angle on a cathode-ray tube. Perhaps the easiest interpreted display is one of a circular trace in which the beam is deflected by the voltage from the pick-up, that is, producing on the circumference of the trace a distinctive mark corresponding to the angle of out-of-balance.

Since the pick-up produces a voltage proportional to vibration velocity, a very simple amplifier can supply an indicating meter suitably scaled in velocity units. The instrument is very much in a state of development, and it may be some time before it can be tried out on a turbine installation.

MICROPHONE CALIBRATIONS

This section deals with the calibrations of microphones used for noise measurements. Microphones used for marine engineering purposes are becoming commonplace, and it is necessary to look into their behavior and performance if reasonably accurate measurements are to be undertaken. The requirements of a microphone for use in typical marine installations are rather more difficult to achieve than those encountered in domestic situations. During 1949 Pametrada began building suitable apparatus for calibrating microphones, so that an accurate check could be maintained and any inconsistencies due to their working conditions taken into account.

The microphones, with associated apparatus, are used to measure or analyze the noise produced by marine turbines, auxiliaries, etc., and may take place both ashore and afloat. When used, particularly at sea, they are inevitably subjected to arduous physical conditions of high temperatures, atmospheres having a high content of oil and salt, and water vapor. Quite frequently they are subjected to rough handling which increases the possibility of permanent changes in sensitivity. It has been found that microphones of a crystal type have eventually failed, and previous to failure the sensitivity was doubtful and unstable. It should not be assumed that the original sensitivity will be maintained throughout the working life of the microphone. They often age due to fatigue, sometimes at a negligibly slow rate, but the importance of regular calibration ensures that the optimum accuracy is realized.

The advantages to be gained by having at the Association a readily available microphone check do not need to be stressed unduly: it is only necessary to point out that whilst national organizations could carry out the measurements, there is always the possibility that the microphone is not available when required for a trial. Similarly, any special arrangements of associated amplifier networks and feeder cables arranged for a particular test can be completely checked *in situ*, and overall corrections given.

Type of Calibration. It is necessary to check the microphone over as wide an audio-frequency range as possible, from 100 c/s to approximately 10 kc/s, and at sound pressures as high as 100 db., with an overall accuracy ± 1 db. The design therefore has been evolved with these characteristics in mind, and is one of obtaining a sound field which is freely progressive, that is, without echoes. The audio range is divided between two especially designed acoustic chambers. The first accommodates

the low frequencies, 100 to 350 c/s, and takes place in a duct; from 300 c/s to about 10 kc/s, or more, a small room is used.

The calibrations are essentially suitable for pressure-actuated microphones and are in terms of the open-circuit e.m.f. of the microphone as compared with a standard accuracy microphone, and are therefore ultimately referred to a reference level of 0.0002 dynes per sq. cm. In practice it is not always possible to obtain the ideal free-field characteristic for a variety of reasons; the source and detector are not points, the source, usually a loud-speaker, has many deficiencies, its polar diagram restricts its use at high frequencies to the axis, nor does it vibrate uniformly at other frequencies.

The Pametrada chamber has internal dimensions of 4×4×4 ft. at the wedge tips. The wedges are of resin-bonded fibreglass and line all the surfaces, and give a lower cut-off frequency in the region of 300 c/s. The equivalent surface area of all wedges (216) is 290 sq. ft. and a microphone at 2 ft. would be subjected to a reflection of 7.4 per cent if the average coefficient of absorption was 0.9. A room is satisfactory if the reflections are less than 10 per cent, about 1 db. from the true characteristics. The characteristics of the Pametrada chamber were measured at several frequencies and shown on Fig. 5. Ambient noise levels of 40 db. are registered inside the chamber when the noise outside from the adjacent boiler house is +100 db. The calibration of microphones below 350 c/s is not suited to the chamber, because of the approach of the cut-off frequency of the wedges and the consequent increase of reflected energy, and the microphone is transferred to a low-

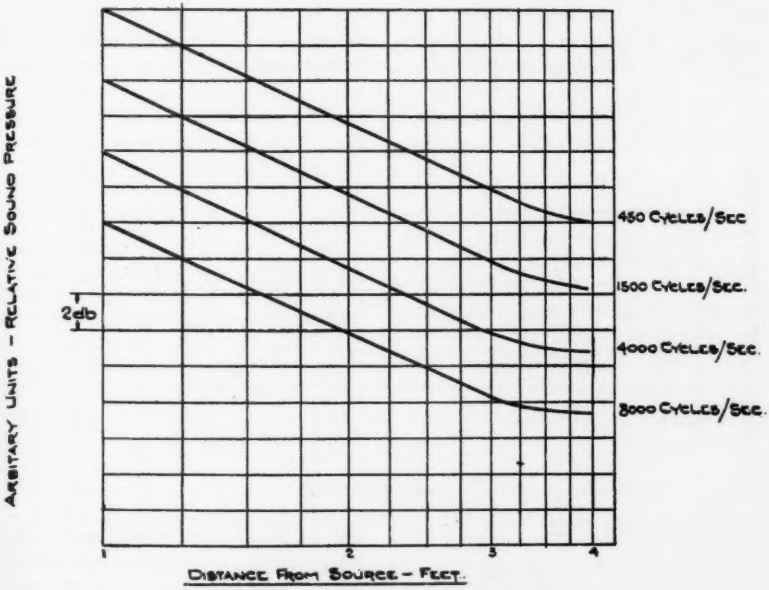

Fig. 5—Characteristics of anechoic chamber.

frequency duct. Essentially, the duct is a lamination of one section of the chamber, but to a larger scale. The absorbing section is tapered at one end of the duct and its length corresponds to half-wave length of the lowest frequency to be absorbed, in this instance a wedge 10 ft. in length would absorb down to 110 c/s; in practice it is usable down to 50 c/s.

As the section of the duct is 12×12 in. a limit must be placed on the upper frequency at a theoretical value of 550 c/s. At this frequency a cross-mode resonance occurs either in the horizontal or vertical plane. Positioning of the microphones in a longitudinal or any other direction is not critical, since a movement of several inches is small compared to half a wave-length. The free-field characteristic is shown in Fig. 6.

The electronic amplifiers, which are necessary for measuring the voltage generated by the microphones, are supported by a 19-in. rack 6 ft. in height. The rack houses an audio-frequency oscillator and power-amplifier which energizes either the loud-speaker in the duct or the loud-speaker in the chamber, and amplifier channels for the "standard" microphone and "test" microphone with their respective decibel meters. Both channels are similar but differences do occur with their input circuits to ensure that the microphones are adequately terminated. A sound pressure is established in the chamber at a particular frequency, with the standard microphone at a convenient distance from the loud-speaker. The attenuators are adjusted until a reading is obtained on the decibel meter, and this is noted. The standard microphone is removed and the test microphone is substituted in exactly the same position, that is, the distance of its diaphragm to the speaker is the same, the same pressure is re-established and the attenuator settings again noted. The difference between the two readings is the difference in sensivities of the two microphones and as the true sensitivity of the standard microphone is known this

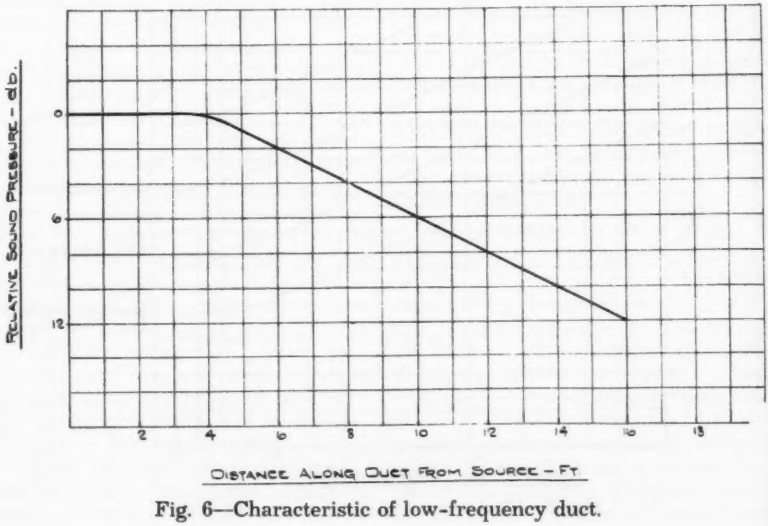

Fig. 6—Characteristic of low-frequency duct.

is added to, or subtracted from, the difference readings and gives the sensitivity of the test microphone. At higher audio-frequencies, above about 2,000 c/s, measurements of the sound-pressure are normally made at several incremental distances from the loud-speaker and a mean value is obtained.

Time alone will indicate what improvements or additions may be necessary, for instance, other means for calibration at extremely high sound-levels of 130 db., and the subject of under-water sound calibrations has not, as yet, been directly considered.

THE SELECTION OF MICROPHONES FOR MARINE USE

Microphones may be classified in a relatively easy manner depending on the method used in converting acoustical energy into the electrical form. This system is shown in Fig. 7, in which they are grouped according to whether they function as capacitors, resistors, or moving-conductor types, etc. From this beginning it is evident which microphones are passive and which are active, that is, those which require some additional external excitation before they will produce an output voltage, and those which do not. For instance, a piezo-electric microphone will produce the required output voltage without further stimulus, but the condenser microphone requires polarization from an external voltage source before it will deliver its output voltage, even though they are both capacitor types.

Generally, most of the microphone types classified are not to be recommended as suitable for marine noise investigations. Among those types which are usable would be the moving-coil, condenser, and piezo-electric crystal. All three have excellent frequency-response characteristics, and whilst there are differences in sensitivity and electrical properties which may involve different electronic amplifiers, they are very well suited to the rigors likely to be encountered.

It is perhaps significant that whilst a first-class moving-coil or condenser microphone is practically stable from atmospheric influences, the piezo-electric microphone does suffer if exposed to high temperatures and humidities. The temperature effects result in the complete destruction of the crystal if subjected to temperatures of the order 130°F. The maximum usable temperature should be limited to about 112°F. At Pametrada a marine version of a moving-coil microphone has been in use some 2½ years without any detectable change in performance, and appears, so far, to be well suited to the purpose.

Concuding, it is our practice always to use moving-coil microphones whenever a large scale investigation is to be carried out. When it is necessary to use commercial instruments employing piezo-electric microphones a frequent check is made. Most of these instruments are, however, portable, and that in itself limits their use in difficult conditions.

TORQUE-METERS

Most machinery trials require the measurement of torques. A range of 9 torque-meters was envisaged, from 500 to 30,000 lb.-ft. with an accuracy of ±½ per cent. These were all to be of the magnetic strain-gage type with no slip-rings, using toothed rings secured to the shaft by non-magnetic

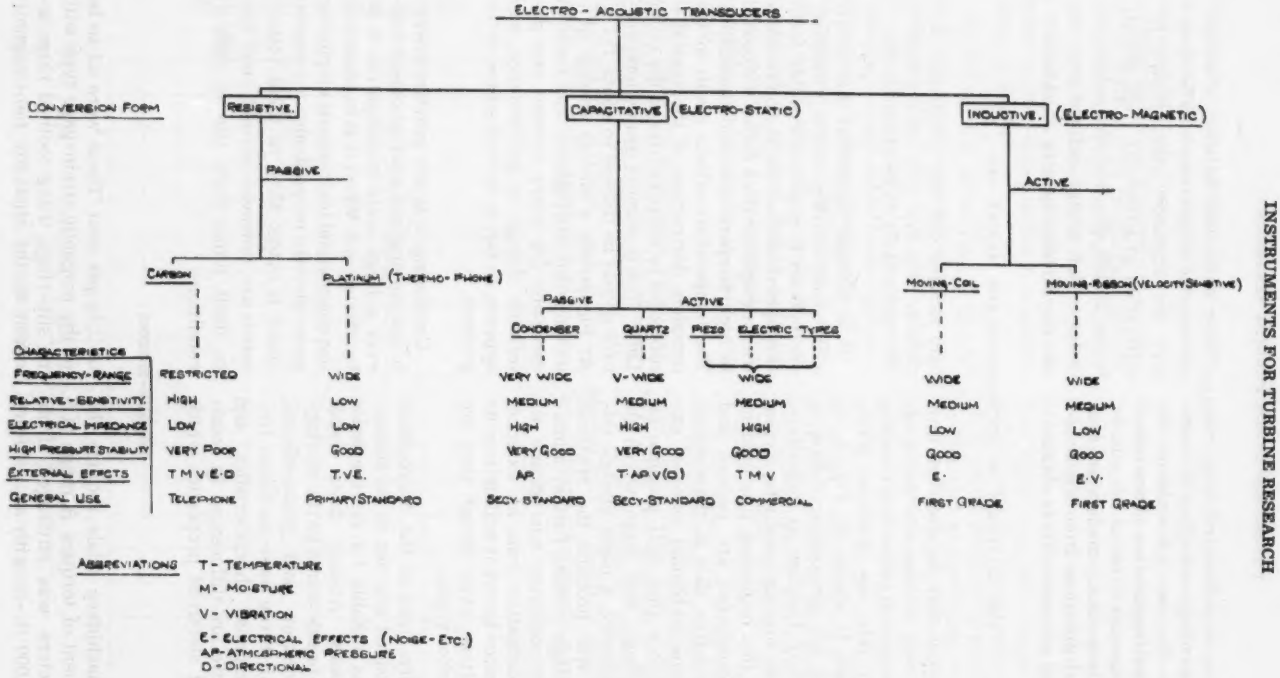

Fig. 7—Properties of various types of microphone.

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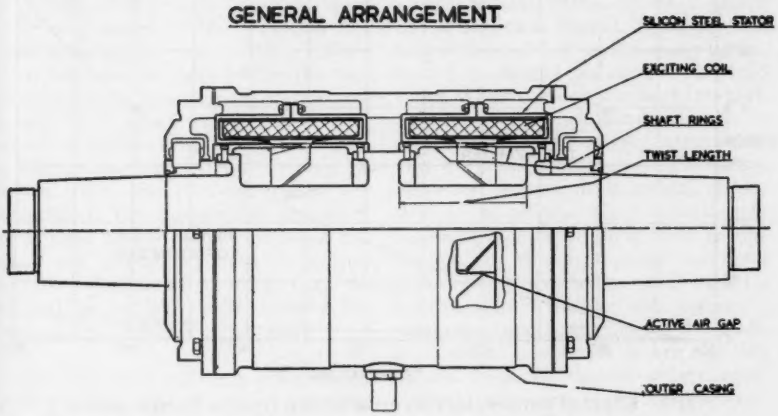

Fig. 8—Electromagnetic torque-meter arrangement.

spacers.[4] The arrangement is shown in Fig. 8. Two active gaps are used, one closing and one opening as torque is applied to the shaft; flux is fed to the active gaps through static radial gaps. Freedom from slip-rings is an important advantage, and the torque-meter can be designed so that bending, axial and tilting movements cause negligible changes in calibration. Moreover, since operation is independent of speed, static calibration may be used. The first torque-meter had soft iron rings and was excited with 100 volts 50 c/s A.C. It was clear that the apparent advantage of null indication could not be realized in practice, and a direct-reading instrument was designed. It had two meters, one reading approximate torque to within a few per cent, and the other expanding any fifth of the approximate scale (selected by a switch) and reading accurately. The accuracy was about ¼ per cent full-scale. A large torque-meter—500,000 lb.-ft.—for a low-speed shaft was arranged for a different type of indicator. The torque-meter coils were arranged in a bridge whose output was measured directly with an 8-in. "Cir-scale" instrument. The only controls necessary were thereby reduced to a supply voltage adjuster for initial setting-up. The coils were split horizontally.

Developments along these lines were made as difficulties appeared, and fairly large zero drift was reduced by attention to mechanical construction. Detailed work on the properties of the magnetic circuit used in torque-meters was completed some time ago and a new instrument showing a beryllium bronze shaft and a thoroughly annealed silicon iron magnetic circuit was made. Its performance as regards sensitivity, stability and power loss in the iron is greatly in advance of its predecessors; the reduction in temperature error is shown in Fig. 9. Beryllium bronze has several valuable properties as a torque-meter shaft material. Its stability and ultimate tensile strength are high, and the modulus of elasticity is such that a much larger shaft twist may be obtained than for steel, dimensions and maximum stress being equal. Hysteresis in the material is low, though it has been shown that

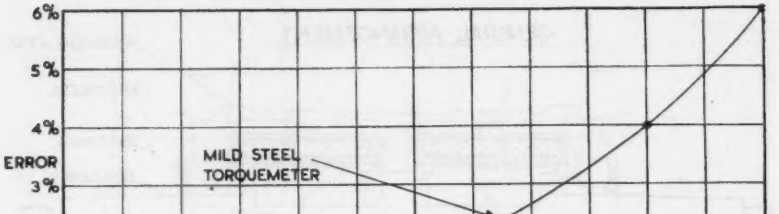

Fig. 9—Effect of temperature on error of two types of torque-meter.

the hysteresis effects in torque-meters, which result in a shift in zero after change of load, are due principally to the material in the magnetic circuit and not to the mechanical properties of the shaft. The cost and greater difficulties of working beryllium bronze probably do not justify its use in ordinary torque-meters. This torque-meter is arranged with the stator hanging on the shaft so that relative movement is minimized and installation facilitated. The indicator is a moving coil instrument directly connected to the stator coils in a bridge circuit, valve amplifiers being eliminated.

This torque-meter measures torques up to 5,000 lb.-ft. at speeds up to 6,000 r.p.m. with an error of less than ± 1 per cent. over most of the range. Below 10 per cent. full-load torque the errors are slightly greater. It is not affected by changes of supply frequency or voltage. This type of instrument is suitable for high-speed shafts up to about 50,000 lb.-ft. but it has become clear that the design is not suitable for low-speed high-torque shafts. In the first place it would be difficult to fit to existing shafts and would have to work with very low shaft stresses. In addition, the very large magnetic circuit would have large losses and uncontrollable leakages. Torque-meters which are not electrical are being investigated for this range of measurement.

An entirely different approach to the problem is provided by what may be called time-interval torque-meters. If a mark is made at each end of the twist length of a shaft, and suitable fixed pointers arranged to coincide with them at zero torque, the twist of the shaft under load may be measured in terms of the displacement of the marks. If one mark is held in coincidence with its pointer, the displacement of the other is a linear function of torque. There are several approaches to measurement in practice. One is to attach an alternator to each end of the twist length and measure the phase difference of the generated voltages. It was decided to use a different system to measure torque in a back-to-back gear test, where an approximate measurement of torque was required and a suitable electro-magnetic torquemeter was not available. A steel pin was screwed into the shaft at each end of the twist length and fixed magnetic

pick-ups arranged to indicate their time of passage. If the pick-ups are lined up at zero torque, the time difference between the pulses from the two pick-ups is a function of torque and speed. To eliminate the effect of shaft speed, one pick-up may be mounted on a micrometer slide so that it may be moved until the pick-up pulses are coincident. The micrometer measures shaft twist and if the properties of the shaft are known a curve of torque against micrometer reading may be plotted (direct static calibration of the instrument is clearly impossible).

The electronic unit is a coincidence detector. The easiest way of showing coincidence is to display the shaft pulses on a cathode-ray oscillograph. In the gear test for which the equipment was made, the time interval at full load and full speed was 22 microsec.; to achieve accuracy small time intervals had to be clearly displayed. Since magnetic pick-ups give pulses of poor shape, pulse-standardizing circuits are provided for each pick-up. Each delivers a square pulse with a fast edge which is used to trigger a phantastron with a run-down time of about 10 microsec. The triangular anode wave-forms are displayed on a double-beam oscillograph, one triangle being inverted. The time base is triggered from the cathode of the driving-end phantastron and the sweep time may be shortened as the micrometer is used to bring the two pulses into coincidence. Using the fastest sweep the sloping sides of the triangles are adjusted to intersect on the horizontal centre-line of the trace. A beam brightening pulse is taken from the screen grid of one phantastron.

In principle the arrangement is quite good, and performs well in the laboratory. On a standard oscillograph the brightness is rather low because the pulses appear for only, say, 20 microsec. in 10 millisec., or for 1/500 of the time (this is at 6,000 r.p.m.: at lower speeds the indication is worse). In the laboratory a twist of 0.01° could easily be detected, or about 1 per cent. of full-load twist. When used on the back-to-back test, however, the difficulties came to light. Some were purely mechanical and were easily overcome. There are, however, several fundamental objections to the scheme which were largely foreseen, and were confirmed in operation. In the first place the fixed pick-ups by which the shaft twist is measured must be fixed with respect to each other and to the axis of the shaft. It is almost impossible to arrange this to the required degree of accuracy. Various arrangements of spaced pick-ups may be used to compensate for deflection and tilting of the shaft, but vibration of frame or floor girders leads to bad jitter of the pick-up pulses. Apart from this, electromagnetic and electro-static pick-ups give a poor pulse shape which has no well-marked reference point. More promising results could be obtained from pick-ups capable of giving a fast edge on their output wave-form. At the moment no further work is being done on these lines.

A very elementary and useful torque-meter was put together for a disc tester. The twist length of the shaft is a helical or spiral spring (both have been used) to one end of which is attached a circular graduated scale, and to the other a pointer which is carried over to read against the scale. It is adjusted to zero with the rig stationary. A cam on the shaft closes a pair of contacts to trigger a flash tube, which is the main illuminant of the scale. This stroboscope enables the twist to be read with quite high enough accuracy for the purpose, and the instrument is cheap and quickly put together. It suggests some improvements in the time-interval torque-meter mentioned above.

Some development work has been

done on an integrating torque-meter which is the subject of a patent application. It would read work done by a shaft. This instrument requires a marker pick-up at each end of the twist length, as in the time-interval torquemeter, arranged to produce coincident pulses from index pins at zero torque. On the shaft is a toothed wheel inducing pulses in a third pick-up. When the shaft is loaded the time interval, t, between the index pulses is proportional to torque T, and inversely proportional to shaft speed, S; the frequency, f, of the pulse train in the third pick-up is proportional to shaft speed. If a number of pulses, N, which occur between the two index pulses be counted then

$$N = K_1 ft.$$

now $t = K_2 T/S$ and $f = K_3 S$

$$\therefore \quad N = K_4 T.$$

If the groups of pulses are added for unit time, the sum

$$N_2 = K_4 ST$$

is proportional to power, and for any other period to total work done. The twist length used could be, say, a tail shaft, to obtain a large twist, but study of a typical example shows that a layout as simple as that described above is not practicable if good accuracy is required. Various expedients may be adopted to improve it.

TORSIONAL VIBRATION RECORDER

Another instrument uses a toothed wheel and magnetic pick-up combination to give a pulse train of frequency proportional to shaft speed. This is a torsional vibration recorder which gives output on a standard pen-recorder. The generated pulse train has its repetition frequency modulated by torsional oscillation of the shaft and by changes in engine speed. It was required to demodulate the pulse train and feed the torsional oscillation component to a pen-recorder. At the same time, indication was required of the steadiness of engine speed so that records could be taken when all variations at the measuring position were due to the torsionals under investigation, uncomplicated by variations in engine speed. The transducer consists of a toothed wheel, fastened to a stub shaft on the end of the shaft under observation. Overhung on the wheel is a support for the magnetic pick-up, which is prevented from rotating by an arm clamped to the machine.

The specification called for a maximum pulse repetition frequency of 10 kc/s and a torsional oscillation frequency range of 1-15 c/s. Four channels were to be provided so that records could be taken on all 4 pens of the recorder simultaneously. Two meters were to show engine speed and rate of change of speed, to be switched as required into each of the four channels. The measuring circuit is a frequency meter consisting of a diode feeding a resistance-capacitance smoothing circuit. The input to the diode circuit is obtained by squaring and limiting the pulses from the toothed wheel pick-up so that the input pulses are all identical in amplitude and shape. The pulse-recurrence-frequency and circuit time-constants are such that variations in output voltage proportional to variations in shaft speed are superimposed on the direct voltage proportional to mean speed. The direct voltage is metered via a cathode follower; the variations operate the pen-recorder. To indicate slow changes of speed due to changes in engine speed a double-triode circuit measures rate-of-change of output voltage, and shows per cent. speed change per sec. on a centre-zero instrument. Torsional oscillations appear on this meter if slow enough for

the meter movement to follow, but give equal excursions either side of zero if mean speed is constant. This meter is used simply to indicate when engine speed is sufficiently steady for a record to be taken. The speed meter is calibrated in "per cent. rev. per min." and controls are provided to set the meter to 100 when the engine is running at a convenient nominal full speed, say, 100 or 200 r.p.m.

The equipment has given satisfactory results on several trials and only detailed modifications have been made in the light of experience. A mechanical calibrator has been constructed; it consists of a motor-driven toothed wheel running at constant speed, torsional oscillations being simulated by mounting the pick-up on a motor-driven rocker arm so that it oscillates with known amplitude and frequency at the periphery of the wheel. The speeds of both motors may be varied so that calibration runs may be made over the whole range of the instrument.

TEMPERATURE-SENSITIVE DEVICES

The measurement of temperatures in marine engineering research requires the use not only of the well-established techniques of thermocouples, or mercury-in-glass thermometers, but also the investigation of newer devices. In those conditions requiring the measurement at temperatures above about 450°F. the thermocouple is almost universally employed, but below this temperature the use of glass thermometers is not always satisfactory. This is often so when it is necessary to make detailed measurements on individual components embodying some 50 measuring points, for example, the temperature distribution of bearings or the temperature distribution of steam condensers.

Investigations were carried out on a material which acts in the manner of a resistance thermometer and seems suited to these latter types of measurements. The material is marketed commercially under the name "Thermistor," and has the property of a rapidly decreasing resistivity with increasing temperature, and is one form of a semiconductor. The small flow of current through the thermistor is therefore an indication of temperature, and it is sufficient to operate a robust meter. As the sensitivity is not absolute it is necessary to calibrate each thermistor, and as will be noted its characteristics is not linear, but this can be improved to a great extent by externally connected resistors. Typical installations of thermistors have included: steam condensers; oil-fuel heaters; gearbox bearings; and air temperatures in compressor systems.

Generally indications of the temperature can be given to better than 1°F., and depending upon the type of thermistor and range involved accuracies to ¼°F. can be obtained, as was the case of the temperature differences across the condensers. Thermistors have enabled accurate temperature measurements to be taken on apparatus which would be otherwise difficult if conventional means had been used, and whilst it is realized that the applications are specialized, further experience will show if these newer techniques are suitable for everyday engineering use. The advantages of the thermistor can be summed up as follows:

Sensitivity. Compared with a chromel-alumel couple the thermistor has a sensitivity of about 4 mV. per °C. as against 0.04 mV. per °C., at the lower temperatures.

Cabling. No special "compensating cables" are required, ordinary copper cables being all that is necessary, providing of course their resistance is not comparable with the thermistors' resistance, that is, at the highest temperatures.

Instrumentation and Supplies. Special instruments are not required, standard milli-ammeters up to 6 in. in diam. are often used. The thermistor being an electronic conductor it can be used with A.C. or D.C. supplies; D.C. is preferable and more conveniently obtained from small batteries.

The disadvantage of the thermistor, as far as marine installations are concerned may be listed as follows:

Calibration. Each one must be calibrated, as the resistance/temperature characteristic is not identical for each thermistor. The calibration should be done in any event to ensure that the inherent time-lag due to its protective mounting has been taken into account.

Linearity. Basically they are non-linear devices but the scale shape may be linearized by a combination of series and shunt resistors for a given range of temperature.

Mechanical. The smaller units are in themselves fragile, and careful design of suitable mounting plugs is required if reliable operation is to be achieved. The larger units are more robust, but since the majority of styles available are enclosed in glass, special protection is required.

The use of the thermistor indicates a method developed for measuring relatively steady temperatures; the following device is being developed for the measurement of rapidly changing temperatures.

In the first instance a high-speed temperature recorder has been made for the purpose of studying the properties of turbine gland strip materials. The material forming the gland-strip is subjected to a known controlled amount of contact, and dispersed along the material are up to 24 thermo-couples. During the period of contact, about ½ sec., the temperature rise produced throughout the material is recorded in the following way. The connections from the thermo-couples are taken to the stator studs of a rotary switch, the selection of the separate thermo-couples being effected by the rotor contact. A second switch identical with the first is connected across tapped resistors across which a D.C. voltage exists. This latter switch when operated is connected to the X-plates of an oscilloscope and causes the beam to deflect horizontally in 25 equally spaced steps, that is, each position of the beam represents one particular thermo-couple. The beam at each position is deflected vertically to a height proportional to the temperature existing at the thermo-couple. The thermo-couple voltage is applied, after being selected, to a D.C. amplifier and then to the oscilloscope amplifiers thence to the Y-plates.

The switch rotor contacts are rotated at 250 r.p.m., each thermo-couple is therefore scanned 4 times per sec., and each displayed for 1/200 sec. By this means an overall picture is built up on the screen showing the temperature distribution throughout the gland material. A permanent record is taken of the temperatures by means of a camera synchronized with the application of the rubbing force. In order that the calibration of the record can be ascertained, one of the thermo-couples is kept at a known temperature, and the sensitivity of the system is such that a 1-in. deflection on the screen is equal to 85°F., the range of the instrument is up to 800-900°F. As the device is relatively new to this kind of application only a little experience has been gained on its limitations. It is clear

that the duty imposed on the rotary switch contact leads to wear and subsequent faulty indication; further use will naturally indicate what modifications will be necessary for more reliable operation.

R.P.M. INDICATOR

This instrument was designed for use on machinery trials which demand very accurate speed measurements. The specified accuracy was ±1 r.p.m. from 500-10,000 r.p.m. It was also required as a sensitive indicator of steady running on trials where this was essential.

Electronic counters are used to obtain the required accuracy. The basic principle has been applied previously, and in other connections. Each revolution of the shaft is signalled by an electrical pulse from a pick-up or tachometer generator. The pulses are counted electronically for an accurately measured time and the result displayed on a dial, or by illuminated figures. The counting period is usually derived from a crystal oscillator or tuning fork, whose output frequency is divided electronically to give a pulse train at comparatively low frequency. Two successive pulses may be used to mark the beginning and end of the counting period. The counters operate well within their maximum counting rate, and count with no error. Overall errors in the instrument are due, therefore, to inaccuracies in fork frequency and to the fact that information is obtained from the shaft only at discrete intervals.

Known instruments[8] gave spot readings of shaft speed on pressing a button, but the requirements mentioned above implied an instrument to count speed, display the count and automatically begin a new one. A continuous reading of almost "instantaneous" speed would therefore be obtained. The usual indication in instruments of the type is by providing a lamp for each decimal digit, the appropriate ones being illuminated. In the present instance the reading was to change at frequent intervals, and to be read from various positions round the machinery, so improvement in indication was necessary. The display adopted was one in which the figures were shown on lamp mosaics like theatre program number indicators.

The operating sequence must provide for making the count, storing the result, resetting the counters to zero ready for the next count, and changing the displayed figures at the correct instants. It was decided to count for 2 sec., which is about the shortest time possible if the display is not to change confusingly rapidly. At the end of the count the result is transferred to a gas-triode store which operates relays to select the appropriate lamps in the mosaic. As soon as the indication has changed the counters may be reset to zero and a new count begun. The store, meanwhile, remembers the previous count until a cancelling signal is received; this cancelling pulse extinguishes the gas triodes and allows those primed by the counters to fire when voltage is restored. This type of display involves a good deal of complication and several circuits were investigated before a satisfactory one was obtained. Cold-cathode valves are used, and each relay operating coil is shunted by a germanium diode to prevent restriking. Since full-scale reading is 10,000 r.p.m. and the count lasts for 2 sec., the maximum counting rate is 5,000 per sec. This is not very high, but with an eye to extensions of measuring range and other uses, hard valve counters were used, identical in

each decade. The decades are separate units with plug-in connections so that a spare is always available in the event of failure. A 1,000 c/s tuning fork is used as time standard, because low-frequency crystals were not easily available when the design started. The sequence controller arranges the following operations:

$t=0$ begin count
$t=2$ stop count, reset indicator
$t=2\frac{1}{4}$ reset counters to zero
$t=2\frac{1}{2}$ begin new count.

A block diagram of the instrument is given in Fig. 10. The first step is to obtain a pulse train with a ¼-sec. period. This is done with a chain of three phantastron frequency dividers, dividing by $10\times5\times5$. From this ¼-sec. pulse train are derived the signals which control the operating sequence of the instrument. The pulses are fed to a scale-of-ten counter from which pulses are taken at counts of 8, 9 and 10. These pulses occur 2, 2¼ and 2½ sec. after the beginning of the count and are used to initiate the processes detailed above. The pulses obtained from the machine shaft are applied to a gate which is opened at instant 0 and closed at a count of 8. The same pulse resets the store so that the new reading is displayed. At a count of 9 a relay opens the grid returns of all the counter valves which are "on" in the zero state so that they are reset ready for the opening of the gate at a count of 10. This cycle automatically repeats.

Input to the instrument is obtained from a toothed wheel and magnetic pick-up on the engine shaft. With a 2-sec. counting period, a result in r.p.m. requires a wheel with 30 teeth, to give the necessary 30 pulses per revolution. The instrument is mounted in a 6-ft. standard rack and consumes about 650 VA. at 250 volts, 50 c/s A.C. The only control is a switch which selects whether the device measures r.p.m. over 2 sec. or frequency over one sec. It also provides an overall check of circuit operation.

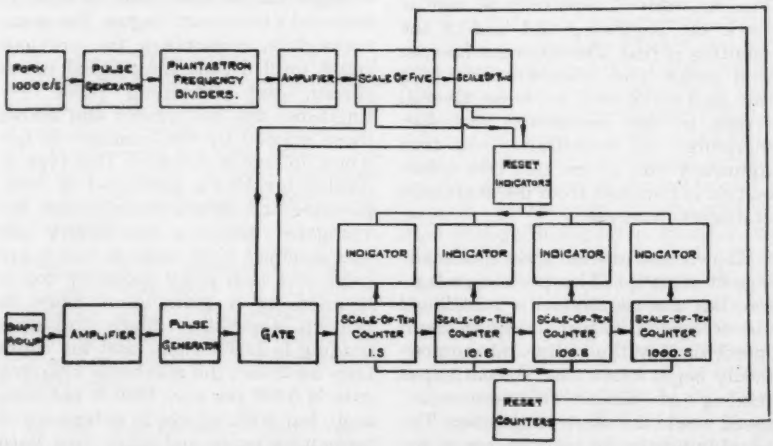

Fig. 10—R.p.m. indicator: circuit arrangement.

FACILITIES FOR ACCURATE CALIBRATIONS

In order to ensure that the accuracy of measurement is maintained within close limits, precision apparatus has been installed, and is mainly for calibrating thermo-couple apparatus, and providing stable audio frequencies which can be used for a variety of purposes. Other accurate references are available elsewhere in the Research Station but as they do not come within the broad activities of the Electronics Laboratory they will be omitted.

The measurement and recording of temperatures is naturally a very important function in turbine machinery research, and to obtain the highest accuracy the apparatus should be referred to precision standards.

The means of doing this is to measure the e.m.f. produced by precision thermo-couples at certain fixed points by a potentiometer, which is essentially free from thermo-electric effects. The precision thermo-couples are usually platinum-platinum 13 per cent rhodium and have a known error of $\pm\frac{1}{3}$°C. up to 1,100°C., other substandards are available in the form of chromel-alumel couples.

Generally the procedure is to determine by means of these thermo-couples the following freezing points:

(*a*)	Distilled water ice	32°F.
(*b*)	Tin	449.4°F.
(*c*)	Zinc	787.1°F.
(*d*)	Aluminum	1,433.8°F.

In addition to the freezing points, calibrations are also determined at the boiling points of water and sulphur (832°F. at 29.92 in. Hg.).

From these known points other thermo-couples can be produced which are in a more suitable form for use with machinery. The instruments normally associated with thermo-couples comprise: multi-point printing recorders, manually-operated potentiometers and indicating pyrometer instruments and their individual calibration can be readily ascertained by comparison to the precision potentiometer. Although this description is necessarily brief, the techniques are well known and space does not permit of more discussion on the procedure and precautions which must be taken to achieve the best results.

So much of the development of electronic apparatus, and its subsequent use in service, relies on oscillators and other timing devices that it became necessary to provide a frequency source whose stability was of a very high order. Generally the indication of frequency of ordinary laboratory oscillators is quite adequate for measurements on sine-wave amplifiers, but is often not good enough for other measurements, such as may occur in the analysis of noise and vibration waveforms, or in the calibration of speed indicating devices.

The Frequency Standard produces fixed frequencies of a sine wave nature at 20, 10, 5, 1 kc/s and 100 c/s. Initially the frequencies are derived from the B.B.C. transmission on 200 kc/s and after amplification and amplitude limitation to reduce the effects of modulation changes, is applied to regenerative dividers. The sine-wave dividers are operated in the following sequence:

200 kc/s	divided	to	20 kc/s.
200 kc/s	"	"	10 kc/s.
20 kc/s	"	"	5 kc/s.
10 kc/s	"	"	1 kc/s.
1 kc/s	"	"	0.1 kc/s.

The outputs of the dividers are connected to suitable transformers and feed distribution cables for supplying various sections throughout Pametrada. Generally the accuracy of the derived frequencies, which is of the same order as the input, is within a few parts in 10^8.

CONCLUSION

The apparatus described in this paper indicates the techniques being used at the moment, and the philosophy underlying their use. After developing equipment to give readings and analysis "at sight," the next logical step is to provide fully automatic operation. The wave correlator is a well-tried automatic means of analysis, and an automatic wave analyzer has been briefly mentioned. It is usually convenient to modify electronic instruments to provide signals to initiate the operation of several instruments in a chain; thus several measurements on the same rig may be synchronized, and, if necessary, recorded on the same tape or film. Future instruments may well benefit from these automatic methods.

It will be appreciated that underlying the development of accurate and reliable instruments is the development of good components. Resistors, capacitors and valves are cheap and plentiful only because the radio industry provides a huge market. Great care must, however, be taken in selecting components from the large variety offered, because those which are good enough for the domestic radio receiver will not necessarily do for precise instruments used in a hot, damp atmosphere and subjected to heavy vibration. The flexibility conferred by plug-and-socket connections is dearly bought if reliability suffers from cable breaks at the plugs. Valuable guides to good practice are provided by the G.P.O. and Service Departments, who use electronic equipment under arduous conditions. However, the unique conditions of marine turbine research work present problems for which the proper solutions must be found by anticipation and experience.

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A PRACTICAL METHOD FOR CLEANING AND DESCALING CARGO TANKS CONTAMINATED WITH BLACK OIL RESIDUES

J. F. MILLS, F. E. COOK and H. S. PREISER

THE AUTHORS

This article was originally submitted as an appendix to "Electrolytic Descaling," by the same authors, which was published in the November 1954 issue of the JOURNAL.

Mr. Mills is Head of Material Laboratories, Puget Sound Naval Shipyard, Bremerton, Washington.

Mr. Cook is Deptuy Head of Coatings and Preservation Branch, Material Developments Division, Bureau of Ships.

Mr. Preiser is a Corrosion Engineer in the Coatings and Preservation Branch.

More complete biographies of the authors will be found on page 1005 of the November 1954 issue.

EDITOR'S NOTE: *Following this article, beginning on page 208, will be found errata and corrections applicable to the previous article "Electrolytic Descaling" which begun on page 1005 of the November 1954 issue of the* JOURNAL.

The selection of a cleaning technique for the removal of oil contamination and scale from cargo tanks is dependent on many factors. The more important considerations include the extent and nature of the contaminant, extent and degree of scale formation, type of repair or alteration and the use for which the cleaned tanks are intended. Several methods for cleaning tanks are available; each having its own limitations and advantages. Sandblasting or gritblasting, for example, is excellent for scale removal in irregular and poorly accessible tanks but in general the process is noisy, expensive and dirty to handle. Electrolytic descaling, on the other hand, is clean, economical and quiet but it is not readily adaptable to irregular, confined tanks. Large rectangular tanks present no problem when cleaned by the electrolytic process.

In either method of scale removal, it is necessary to gas free the tanks and remove oil residues prior to commencing operations. The decontamination of tanks containing heavy fuel oils can be accomplished by injecting hot seawater, hot oil or chemical solutions through revolving nozzles under pressure (Butterworthing) or by

the vapor injection-emulsion cleaning process. The former cleaning technique, using hot sea water under pressure, often is accomplished at sea. The use of hot oil or chemicals results in a more thorough cleaning job but introduces complications in the handling procedure. Vapor injection-emulsion cleaning is mainly a shipyard technique but it can accomplish difficult oil removal with a minimum of time, effort and expense.

The method described in the paper, developed at the Puget Sound Naval Shipyard, is particularly applicable to the removal of heavy scale formation contaminated with black oil residues. The first phase entails the removal of the black oil residues by means of a vapor injection-emulsion cleaning technique. The second phase which follows, removes the rust and corrosion products by means of the electrolytic descaling process. The laboratory and field investigations which were undertaken to develop practical field data for the electrolytic descaling process are described in the paper published in the November 1954 issue of the JOURNAL (1). The bibliography of the previous paper contains 28 background references on the subject.

In cases where tanks are being converted from heavier oil cargo to aviation gasoline, all pipe lines should either be removed and cleaned by regular cleaning methods in the pipe shop or they should be "rattled," flushed with hot water or chemicals, wiped or otherwise so treated as to insure complete removal of all traces of black oil, diesel oil, water, or kerosene from the inside of the pipe lines. This treatment should be extended not only to suction and discharge piping, but where necessary, to vent piping, steam smothering lines, leaky heating coils, pipe handrails, ladders and any other type of piping or structural joint or connection found within the tank. If the tank piping has a packed expansion joint, the packing should be removed and renewed with clean packing. In other cases, where contamination of cargo is not a problem, the exterior surfaces of the piping within a tank could be cleaned in place.

VAPOR INJECTION—EMULSION CLEANING

The emulsion cleaning of fuel oil tanks by the vapor injection process is based upon the diffusion of finely divided particles of cleaner by the movement of steam to the relatively cold surfaces of the tank where condensation takes place and an emulsion is formed. Preheating by steam lowers the viscosity of diesel oil, which is the solvent component of the emulsion cleaner, and aids in the penetration of the emulsifying agent. Over-steaming, however, should be avoided since it raises the temperature of the side walls too much, thus lowering the condensation rate and decreasing the cleaner efficiency.

Tank Cleaning Procedure

There are six steps in the cleaning of an oil tank by this method, which are as follows:

(a) Ventilate the tank. This is done by a suction blower. The tube is extended to the bottom of the tank through the manhole. The displaced air is replaced by entrance of fresh air through the manhole. The ventilation should be continued until the space is declared safe for entrance.

(b) Remove sludge by pumping. Any type sump pump, that will lift the heavy sludge, is satisfactory for this purpose. The pump should be placed in the tank and the suction hose extended to the lowest point in order to evacuate as much sludge as possible. After the sludge has been removed

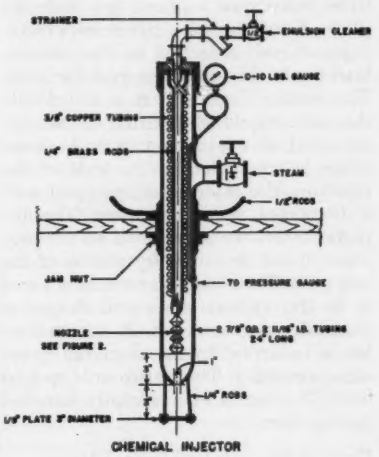

Figure 1

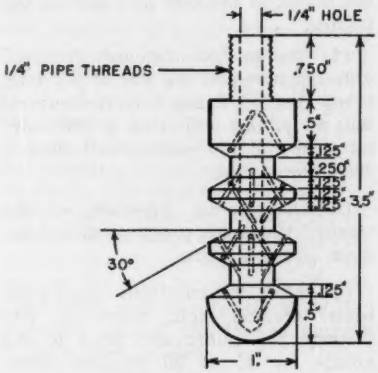

Figure 2

Nozzle

NOTE: Drill 24 1/32" holes, 4 holes 90° on each surface. Holes in opposite surfaces aligned exactly. Three groups of four pair, each group 30° offset.

satisfactorily, the pump and all hoses should be removed from the space.

(c) Open the tank vent to insure against building up a pressure when subsequent steaming takes place.

(d) (1) Figures 1 and 2 show the chemical injector, mounted on a portable manhole cover made of waterproof plywood which has the same dimensions as the regular manhole cover. This assembly is placed on the manhole and bolted down. It should be noted that the lower end of the injector extends below the level of the deck in order to obtain best results. A steam line of 1½" or larger is connected to the valve provided on the chemical injector. The steam pressure on the line should be 100 pounds per square inch.

(2) Close all other openings except the vent, which should be left open at all times, and admit steam at full pressure for one or more hours. The pressure gauge on the injector should show no pressure when steaming. If it does, shut off the steam immediately and check the tank vent. When the temperature, as indicated by the thermometer in the manhole cover, reaches from 175° F to 180° F the tank is ready for the injection of the emulsion cleaner.

(3) A chemical injection pump to supply the emulsion cleaner to the injector must be available. The pump should be a positive displacement type with a capacity of approximately five gallons per minute at one hundred pounds per square inch pressure. The exit side of the pump should be equipped with an adjustable pressure by-pass back to the intake of the pump. The pipe to the nozzle of the injector should be provided with a gauge so that the proper nozzle pressure can be determined and maintained by adjustment of the by-pass. The exit side of the pump should be connected to the chemical injector, the pump started and the emulsion cleaner pumped through the nozzle of the injector. The solution is forced through

the nozzle at the rate of 3 gallons per minute.

(4) The amount of cleaner required will depend upon the size of the tank being cleaned. It has been determined that 50 gallons will clean a 1200-barrel tank and 150 gallons will clean a 6000-barrel tank.

(5) During the injection of the cleaner, the steam pressure should remain on the injector.

(e) After the emulsion cleaner has been injected, cold water is run through the pump and lines to the nozzle for 15 to 30 minutes while steaming. This produces a wet steam and causes excessive sweating on the walls of the tank which emulsifies the oil and washes the emulsion to the bottom of the tank.

(f) As soon as the tank is cool enough to enter, the tank should be washed down with a jet of high pressure fresh water. The sump pump should be lowered into the tank and the suction line placed in the lowest point. The pump should be started and the emulsified oil and water discharged from the tank. If the tank is washed down before it becomes completely cold, no wiping is necessary except in the lowest pockets. Wiping down with rags will be required to remove the final traces of oil which cling to the lowest points of the tank.

Description of Injector

The chemical injector, Figure 1, is designed to preheat and atomize the cleaning solution for uniform mixing with the entering steam. The body of the injector proper, the steam side, is made of copper-nickel tubing to which are connected a pipe coupling, a standard all-thread pipe nipple and three brass rods supporting a deflector plate. A copper coil is silver soldered to support rods attached to the reducer bushing in the top of the injector body. The nozzle, Figure 2, is screwed into the coupling which is fitted to the bottom end of the copper coil. A steam valve is connected to the side of the injector. The injector is equipped with a 10-pound pressure gauge. The injector assembly is mounted on the plywood manhole cover by means of the jam nuts. The covers are manufactured to fit the various sizes and shapes of manholes encountered. A thermometer is mounted in the manhole cover after assembly. The entire unit weighs only 27 pounds and is easily handled by one man.

Formulation of Emulsion Cleaner

The emulsion cleaner is composed of an amine-oleate soap dissolved in diesel oil. The composition of the cleaner is as follows:

2½	gallons oleic acid (red oil)
1	gallon triethanolamine or ½ gallon monoethanolamine
45	gallons diesel oil

In compounding the formula, the oleic acid is dissolved in the diesel oil until a homogeneous solution is formed. Then the amine is added slowly with vigorous stirring in the proportions given above.

This amine soap solution is a liquid at normal temperatures but may gel in very cold weather. It is therefore advisable when compounding the cleaner, to use diesel oil which has been warmed to at least 60° F. Cost of the cleaner is about $30 per 100 gallons. In general, tanks of 50,000-gallon capacity require 50 gallons of cleaner. Tanks of 250,000-gallon capacity require up to 150 gallons of cleaner.

ELECTROLYTIC DESCALING

The electrolytic descaling phase of the tank cleaning method consists of making the inner surface of the tank the cathode in an electrical circuit in which suspended fabricated steel grids or curtains are the anode. The tank is

filled with sea water which functions as the electrolyte. When sea water is not available a 1½ percent salt solution may be substituted. Existing data is based on a sodium chloride solution. However, there is no obvious reason why other electrolytes could not be satisfactorily employed. Electrical energy is supplied by one or more low voltage (about 10 volts), high amperage (about 1200 amperes) DC welding generators, so that a resultant current density of about 0.5 to 1.0 ampere per square foot on the tank surfaces is maintained. For all practical purposes, the current passes from the anode through the electrolyte to the cathode. The length of time required to loosen the scale varies somewhat with the thickness of the scale and the current density. The time required is greater with heavier scale and less with greater current density. Best results are obtained with a current density between 0.5 and 0.7 ampere per square foot for the side and top surfaces, and a current density of about one ampere per square foot for the bottom of the tank. The time required for descaling tanks varies from 24 to 48 hours. For example, for tanks of 500,000 gallon capacity, 7 welding generators (each rated at 1200 amps continuous duty) will do the job. One generator would be connected to the anodes for each bulkhead, one to the overhead anodes, and two in parallel to the bottom anodes.

Anodes:

The anodes consist of curtains or grids of ½" to ⅝" diameter steel rods with hooked ends inserted into holes in angle iron framing or over 1" dia-

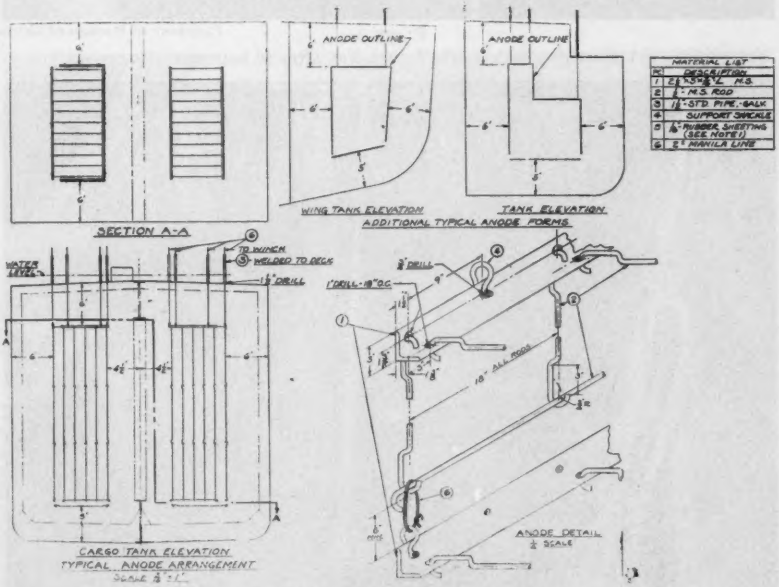

Figure 3

NOTES

1. Where section of anode comes within 1 ft. of ship's structure, shield anode section with rubber, Pc. 5.
2. Omit anode rods where necessary to maintain clearance.

PLATE 1 Courtesy of Bureau of Ships

Suspension of anode grid in tank. View taken after 40 hour descaling period.

PLATE 2 Courtesy of Bureau of Ships

Arrangement of stand pipes on deck for venting and for suspending anodes.

meter rods. All parts of the anode are manufactured in the shop and the individual parts are then lowered through the tank manholes and assembled by hooking and clamping so that a minimum of welding is required in the tank (see Fig. 3). The anodes are then readily disassembled after the tank descaling operation and reused in subsequent tanks until excessively reduced in diameter by electrolytic action. The anodes are so designed and suspended in the tank that all elements are spaced about six feet from each of the six tank surfaces. The rods in the six planes of the anode curtains are spaced on 18″ centers. (See Plate 1.) The grid is supported by a sufficient number of 2″ sisal ropes through 1½″ diameter holes drilled through the deck above the tank to hold the anode in the position previously described. Centered over each 1½″ hole and welded to the deck are pieces of standard 1½″ pipe of sufficient length so that they extend at least one foot above the highest level of the electrolyte in the completely filled tank. The anode supporting ropes will then pass through these pipes. (See Plate 2.) Lashing eyes for these ropes have previously been welded to the top angle iron frame of the anode. In the event that protruding members of the hull come closer than six inches to the anode, that portion of the anode should be protected with split rubber hose. The anode may be manufactured and assembled as a single unit or two or more units may be used in a tank, depending upon the construction of the tank. A deep vertical keel and centerline deck beam may make two separate anodes in a single tank desirable. Also the throwing power of the anodes into corners and recesses can be improved by the use of anode extensions. (See Plate 3.)

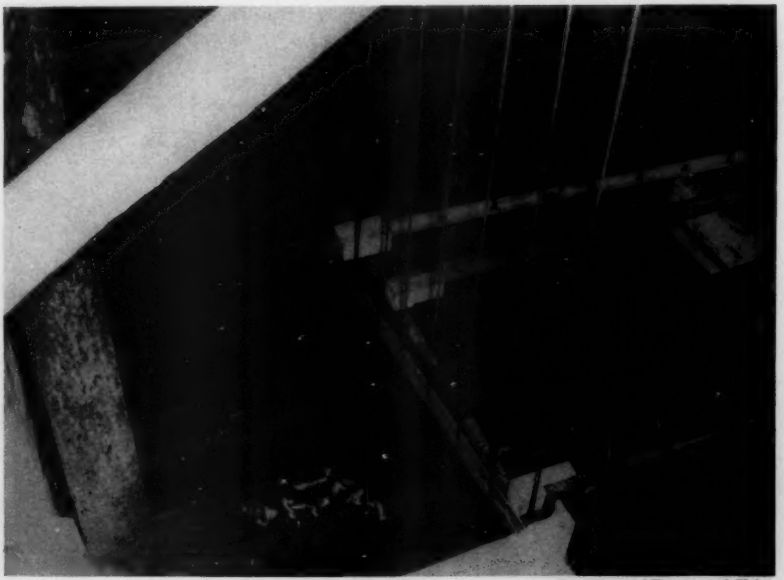

PLATE 3 Courtesy of Bureau of Ships

Grid anodes showing extension rods for corners.

Order of Procedure

After a ship is dry docked ready for cargo tank descaling, the following procedure is followed:

The first step depends upon the condition of the tank, the most recent cargo and whether or not the tank has been hot water washed at sea before coming in for tank cleaning. If the tank has not been hot water washed, and its latest cargo was some type of black fuel oil, the tank is first cleaned as previously described. The tank is then tested again for explosive gases. When safe to do so, an opening is cut in the lowest part of the bottom of the tank for a ten-inch gate valve, which is bolted to the bottom plate. The tank is now ready for the installation of the anodes.

D.C. electrical generator capacity is furnished by a battery of motor-generator sets. (See Plate 4.) These are welding motor-generators with controls modified to make them suitable for the electrolytic descaling service. The modification consists simply of providing separate field excitation from a 70 V.D.C. welding machine connected to each motor-generator to control current output. Maximum continuous rating of the motor-generators is 1200 amperes. The positive leads of these generators are brought into the tank and bolted to the top frame of the anode at points to give the proper distribution of current to the tank surfaces. Electrical jumpers are provided where necessary to aid in the current distribution. The bolted connections and exposed electrical cable lugs are painted with a coating of varnish to insulate them from direct contact with the electrolyte. The negative leads are attached to the shell of the ship with bolted lugs.

PLATE 4 Courtesy of Bureau of Ships

View of motor-generator assembly from vessel.

The tank is now filled with salt water to within 12″ of top of coaming. The hydrogen liberated in the descaling process escapes through the pipes which also house the anode supporting ropes. Current should never be applied with tank only partially full of water. Apply generator load not to exceed 1200 amperes per generator at about 10 volts. Adjust current from 0.5 to 0.7 ampere per square foot of tank interior surface for side and top anode, and one ampere per square foot for bottom anode. Run at this load for 16 hours minimum, drain down, flush out scale taking care to remove all scale accumulations on the bottom of the tank and refill with salt water. Reapply generators and continue descaling for 16 hours (minimum). Reverse the flow of electricity at half the former load for five minutes at the end of the final scaling operation. The tank then is drained completely by opening the 10″ gate valve installed at the bottom of the tank. Flush the entire interior surface of the tank with salt water from the ship's fire mains, or from shore connection, leaving the drain valve open during the flushing operations. Electrolytic descaling is continued if necessary. Otherwise the interior tank surfaces are then hosed down with the fresh water.

Precautions

During the entire operation of descaling, precautions are taken to prevent explosions from hydrogen gas accumulations. Gas test readings are taken frequently in nearby spaces in proximity of the tank. "No Smoking" and "No Welding" signs are posted in the vicinity of the tank.

Where a vessel is to be electrolytically descaled while afloat and generators are located at the pier, the size of copper ground-return cables should be at least 1,000,000 circular mils per 1,000 amperes. This precaution will prevent possible stray current damage to the hull. The electrolytic method of descaling rust from tanker compartments has been found to be quieter, cleaner, quicker, and more economical than other conventional cleaning techniques.

ACKNOWLEDGEMENTS:

The work at the Puget Sound Naval Shipyard was supervised by Mr. J. F. Mills, Head of Laboratory Branch, Code 370. Shop work was performed under the direction of Cdr. A. C. Lombardi, U.S.N., Shop Superintendent. The principle investigators on the various phases of the projects were Mr. W. H. Miles, Test Engineer, Mr. G. F. Pettengill of the Chemical Engineering Unit and Mr. R. E. Canfield of the Chemical Group.

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ELECTROLYTIC DESCALING
AN ELECTRICAL METHOD OF RUST REMOVAL FROM TANKER SHIP COMPARTMENTS

EDITOR'S NOTE: *We seem to have rushed this article to press too rapidly. Special effort was made to include it in the November 1954 issue. In accomplishing this, it was not possible for the authors to read the page proofs to find printer's and editor's errors and to discover methods of improving their own work. The following changes in the article as printed result from proof reading after publication. It is regretted that it is impossible to reprint the corrected article in complete form. It is hoped that each interested holder of the original article will find it not too cumbersome, to merge what follows with it.*

ERRATA AND CORRECTIONS

On page 1008, right hand column, 6th line of second complete paragraph:

"consideration" should be "considerations"

On page 1009, next to bottom line of text:

The word "and" should be inserted between the words "separated" and "suspended"

On page 1010, third line below table:

The word "rubbery" should be "rubber"

On page 1010, second line above the two column narrative:

"fluxural" should be "flexural"

Where appearing — twice on page 1011, once on page 1012 and twice on page 1016:

"Amperes" or "Amps" should be changed to the singular "Ampere" or "Amp"

On page 1012 notes have been added to Figure 4. This figure, as modified is reprinted on page 209.

On page 1013 notes have been added to Figure 5. This figure, as modified, is reprinted on page 210.

On page 1013, Figure 6, the Legend "hours" should be added to the right-hand vertical scale.

On page 1015, last paragraph in right-hand column 4th line:

"incrusted" should be "rusted"

On page 1015, last paragraph in right-hand column, 6th line:

"sanded" should be "sandblasted"

On page 1020, 3rd line from top:

"Alcoa 25" should be "Alcoa 2S"

On page 1023, last line of first paragraph, left-hand column:

"Fi.OH" should be "FeO.OH"

On page 1023, last line in left-hand column:

"move" should be "remove"

On page 1023, 1st paragraph in right-hand column, 6th line:

The first sentence should end with ". . . process." The next sentence should read "The experiments ran concurrently." The next sentence should end in the 8th line with ". . . . cathodes." The following sentence would start "The passage . . ."

On page 1025, 7th line from bottom of left-hand column:

"differencies" should be "differences"

ELECTRODE SPACING VS. POTENTIAL REQUIRED FOR DERUSTING STEEL PLATES
Cathode 5 x 6 x ⅛ inch rusted steel plate anode ¼ inch diameter steel welding rod electrolite 3 per cent sodium chloride solution at 74 degrees F.

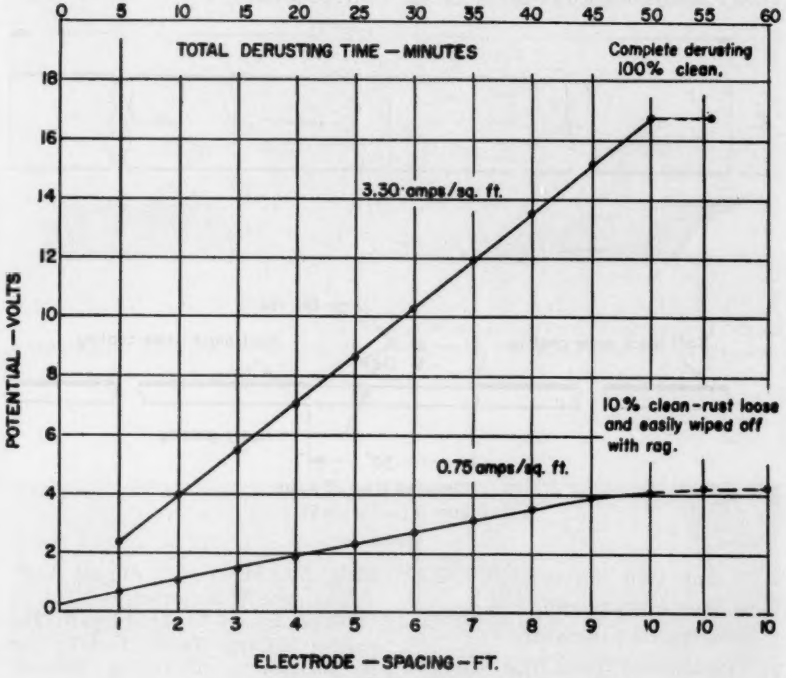

TOTAL DERUSTING TIME — MINUTES
Complete derusting 100% clean.
3.30 amps/sq. ft.
10% clean-rust loose and easily wiped off with rag.
0.75 amps/sq. ft.
POTENTIAL — VOLTS
ELECTRODE — SPACING — FT.

Figure 4

On page 1027, 5th line from bottom of left-hand column:

"25" should be "2S" and "Electro" should be "Electro-"

On page 1029, third line in the legend of Figure 12:

"below" should be "above"

On page 1031, the footnote should be:

"*Adapted from Puget Sound Naval Shipyard 'Electrolytic Descaling' report 222C-3 of November 1949."

On page 1037, 5th line from top of right-hand column:

"of" should be "or"

On page 1037, Section III in three places where it occurs:

"Solomone" should be *"Salamonie"*

On page 1039, under Plate 6:

"Frow" should be "From"

On page 1041, 3rd line of right-hand column:

The word "functions" should be deleted.

On page 1045, 3rd line below Plate 12 in right-hand column the figures have been reversed. It should read "in 24 to 18 hours . . ." instead of "in 18 to 24 hours . . ."

On page 1049, 8th line in the right-hand column should read:

"by Commander A. C. Lombardi, U.S.N., Shop Superintendent"

On page 1050, 1st line, 3rd word:

"Forst" should be "Fort"

THROWING POWER OF A SINGLE ANODE ON RELATIVELY LARGE CATHODE STEEL PLATE
Cathode 6 x 122 x ⅛ inch rusted steel plate. Anode ¼ inch diameter welding rod opposite center of cathode. Electrolyte 3 per cent sodium chloride solution at 71 degrees F. Electrode spacing 6 inches. Cathode current 3.43 amperes.

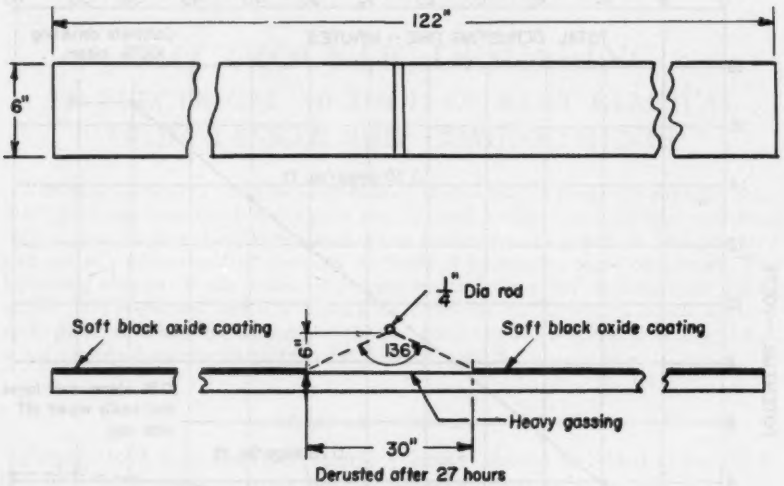


Figure 5 (see table 5)

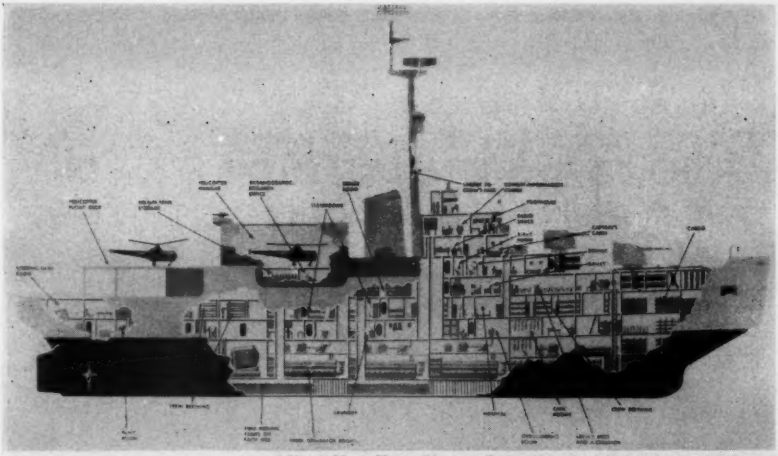
On page 1050, references 26 and 27 have been interchanged.

Reference 26 is correctly:

"Puget Sound Naval Shipyard 'Descaling of Cargo Tanks, Investigation of Electrolytic Method,' Test Report 222-C Supplement 1, 9 August 1948"

Reference 27 is correctly:

"Puget Sound Naval Shipyard 'Descaling of Cargo Tanks, Investigation of Electrolytic Descaling Method,' (NS-062-003), Test Report 222C—Supplement 2, May 1949."



Official U. S. Navy Photo released by the Department of Defense

Artist's cutaway drawing of the USS *Glacier* (AGB-4) The Navy's Largest Ice Breaker.

Bow On View of the Navy's new ice breaker, USS *Glacier* (AGB-4).

Official United States Navy Photograph

Landing Ship, Tank-LST 1163 is one of the new prototypes of another World War II type.

Official United States Navy Photograph

Landing Ship, Dock-LSD 28 is one of the new prototypes of this type landing ship which has proven to be very versatile since its appearance during World War II.

DRIVING POINT ADMITTANCE OF A SYMMETRICALLY FOLDED ANTENNA

COMMANDER CHARLES W. HARRISON, JR., U.S. NAVY

THE AUTHOR

attended the U.S. Naval Academy Preparatory School, the U.S. Coast Guard Academy and the Unversity of Virginia, where in 1939 he received the S.B. degree in Engineering and in 1940 the degree of Electrical Engineer. In 1942 he was graduated with the S.M. degree in Communication Engineering from Cruft Laboratory, Harvard University, and that summer completed the Navy course in Radar Engineering at Massachusetts Institute of Technology. Subsequently for several years he was engaged in lecturing to officers of the Armed Forces assigned to the radar schools at Harvard and Princeton Universities. He has had four tours of duty in the Electronics Design and Development Division, Bureau of Ships; two at the U.S. Naval Research Laboratory; one at the Signal Corps Engineering Laboratories (Evans Signal Laboratory); and one at the Philadelphia Naval Shipyard. His experience includes amateur, naval and broadcast-station operation.

In 1951 Commander Harrison was selected for training in Advanced Science at Harvard University under sponsorship of the Office of Naval Research. This program led to the M.Eng. degree in 1952 and Ph.D. degree in Applied Physics in 1954.

Commander Harrison is presently Electronics Officer on the Staff of Commander Operational Development Force. He is a member of several societies and associations, including the Harvard Chapter of the Society of the Sigma Xi.

This paper was presented at the U.S. Naval Postgraduate School, Monterey, California on 7 April 1954, and at the U.S. Naval Electronics Laboratory, San Diego, California on 12 April 1954.

SUMMARY: *A general method is advanced for determining analytically the driving point admittance of folded wire structures that are symmetrical with respect to the driving point. To illustrate the technique a formula is developed for the input admittance of a three-conductor symmetrically folded antenna having two open-circuited ends. The performance of this antenna and the conventional three-wire folded dipole are investigated numerically and compared. Ancillary tables, having universal application in this field, are supplied to permit rapid numerical studies of the admittance qualities of folded antennas as influenced by changing the dimensions and spacing of the circuit elements.*

I. INTRODUCTION

A valid analytical determination of the driving point admittance of an antenna must proceed from the Maxwell-Lorentz field equations. These equations define the electric and magnetic fields E and B in terms of two universal constants and the density functions (which characterize the av-

erage electrical properties of matter). By specifiying the divergence and curl of a vector potential A and the gradient of a scalar potential ϕ as functions of E and B, the four Maxwell-Lorentz first order partial differential equations may be transformed into two separated wave equations of second order involving only the potential functions. These are known as the equations of d'Alembert. The Helmholtz integrals are explicit solutions of the wave equations, yielding A and ϕ in terms of the density functions. E and B are calculable from A and ϕ by appropriate differentiation. An application of the boundary conditions on E and B permits determination of the distributions of current and charge.

The potentials A and ϕ are defined by the following equations:

$$\text{div } A = -j\frac{\omega}{c^2}\phi \quad \ldots\ldots (1)$$

$$\text{curl } A = B \quad \ldots\ldots (2)$$

$$-\text{grad } \phi = E - j\omega A \quad \ldots\ldots (3)$$

Here c is the velocity of electromagnetic-wave propagation given by the relation

$$c = \frac{1}{\sqrt{\mu_0 \epsilon_0}} = 3 \times 10^8 \text{ meters per second} \quad \ldots\ldots (4)$$

μ_0 is the universal magnetic constant, $\mu_0 = 4\pi \times 10^{-7}$ henry per meter. ϵ_0 is the fundamental electric constant, $\epsilon_0 = 8.85 \times 10^{-12}$ farad per meter. ω is the angular frequency.

$$\omega = 2\pi f = \frac{2\pi}{\lambda} c = \beta c \quad \ldots\ldots (5)$$

In (5) λ is the wavelength.

On eliminating ϕ between (1) and (3)

$$E = -j\frac{c^2}{\omega}\left\{\text{grad div } A + \beta^2 A\right\} \quad \ldots\ldots (6)$$

Equation (6) permits one to calculate, for example, the distribution of vector potential on the surface of a cylindrical conductor of half-length h and radius a oriented parallel to the z-axis of a coordinate system. Let an external driving voltage V_K be applied to a small break in the center of the wire at $z = 0$. The following symmetry conditions obtain:

$$\left.\begin{aligned} A_z(z) &= A_z(-z) \\ I_z(z) &= I_z(-z) \end{aligned}\right\} \quad \ldots\ldots (7)$$

In (7), $I_z(z)$ is the current at any point z along the conductor.

The tangential component of electric field $E_z(z)$ is continuous across a boundary; accordingly, if a perfectly conducting wire is assumed, $E_z(z)$ is zero on its surface. One obtains

$$\frac{d^2A_z(z)}{dz^2} + \beta^2 A_z(z) = 0 \quad \ldots\ldots (8)$$

provided

$$\left.\begin{aligned} a &<< h \\ \beta a &<< 1 \end{aligned}\right\} \quad \ldots\ldots (9)$$

The solution of (8) is

$$A_z(z) = \frac{-j}{c}\left\{K_1 \cos \beta z + K_2 \sin \beta z\right\} \quad \ldots\ldots (10)$$

In order for $A_z(z)$ to satisfy the symmetry condition (7), sin βz must be replaced by sin $\beta/z/$ after K_2 has been evaluated, as explained later.

The integral

$$A_z(z) = \frac{\mu_0}{4\pi} \int_{-h}^{h} I_z(z')\, K_a(z,z')\, dz' \quad \ldots\ldots (11)$$

is a well known solution of d'Alembert's wave equation for the vector potential. It is called the Helmholtz integral. $K_a(z,z')$ has the value

$$K_a(z,z') = \frac{e^{-j\beta R_a}}{R_a} = \frac{e^{-j\beta\sqrt{(z-z')^2+\alpha^2}}}{\sqrt{(z-z')^2+\alpha^2}} \quad \ldots\ldots (12)$$

To compute the vector potential on the surface of the conductor one must specialize (12) by setting $\alpha = a$.

Since the entire circuit consists of one isolated wire (so that there are no contributions to the vector potential due to currents flowing in nearby conductors) one may now equate (10) and (11), obtaining an integral equation in the current $I_z(z)$. Thus,

$$\int_{-h}^{h} I_z(z') K_a(z,z')\, dz' = -j\frac{4\pi}{R_c}\left\{K_1 \cos\beta z + K_2 \sin\beta z\right\} \quad \ldots\ldots (13)$$

when $0 \leq z \leq h$. R_c is the characteristic resistance of space, defined by

$$R_c = \sqrt{\frac{\mu_0}{\epsilon_0}} \approx 120\pi \text{ ohms} \quad \ldots\ldots (14)$$

The constants K_1 and K_2 are evaluated from the boundary conditions

$$I_z(\pm h) = 0 \quad \ldots\ldots (15)$$

and

$$V_K = \operatorname*{Limit}_{z \to 0}\left\{\phi^+(z) - \phi^-(z)\right\}, \quad \ldots\ldots (16)$$

respectively. In (16) $\phi(z)$ is given by (1), i.e.,

$$\phi^+(z) = j\frac{c^2}{\omega}\frac{d}{dz}A_z(z) \quad \ldots\ldots (17)$$

and

$$\phi^-(z) = j\frac{c^2}{\omega}\frac{d}{dz}A_z(-z) \quad \ldots\ldots (18)$$

To evaluate $\phi^+(z)$ and $\phi^-(z)$ one must know $A_z(z)$. This is given by (10). An application of (16) then yields

$$K_2 = \frac{V_K}{2} \quad \ldots\ldots (19)$$

and (13) becomes

$$J_a(z) = \int_{-h}^{h} I_z(z') K_a(z,z')\, dz' = -j\frac{4\pi}{R_c}\left\{K_1 \cos\beta z + \frac{V_K}{2}\sin\beta|z|\right\} \quad \ldots\ldots (20)$$

This expression is valid over the entire interval $-h \leq z \leq h$, because z has been replaced by $|z|$ in the sine term. It is to be observed that (20) satisfies (7). Since K_1 may be determined from boundary condition (15), the only unknown in (20) is $I_z(z)$. It is found by a method of successive approximations.

The driving point admittance is the current $I_z(0)$ divided by V_o. That is

$$Y_a = \frac{1}{Z_a} = \frac{I_z(0)}{V_K} \quad \ldots\ldots (21)$$

The tables given in the Appendix for Z_d and Y_d are the solution of (20), as given by (21), for a symmetrical center driven antenna of half-length h and radius d. These tables also apply to an antenna of radius a. One simply replaces a by d.

II. INTEGRAL EQUATIONS PERTAINING TO A THREE-CONDUCTOR SYMMETRICALLY FOLDED ANTENNA

The folded wire structure to be analyzed in this paper as a boundary value problem in electromagnetic theory is shown in Figure (1). It consists of three highly conducting parallel cylinders (designated (1), (2) and (3)) of half-length h and radius a. The spacing between wires is b, that is the cross-section of the structure, taken at right angles to the wire axes, is an equilateral triangle. Voltage V_o is applied in wire (1) at $z = 0$. Wires (1) and (2) are connected together at $z = h$; wires (1) and (3) at $z = -h$. Conductor (2) is open-circuited at $z = -h$; conductor (3) at $z = h$.

Equation (8) governs the distribution of vector potential along the wires. Inspection of Figure 1 reveals that the following symmetry conditions obtain:

$$\left.\begin{aligned} A_1(z) &= A_1(-z) \\ A_2(z) &= A_3(-z) \\ A_3(z) &= A_2(-z) \end{aligned}\right\} \quad \ldots\ldots (22)$$

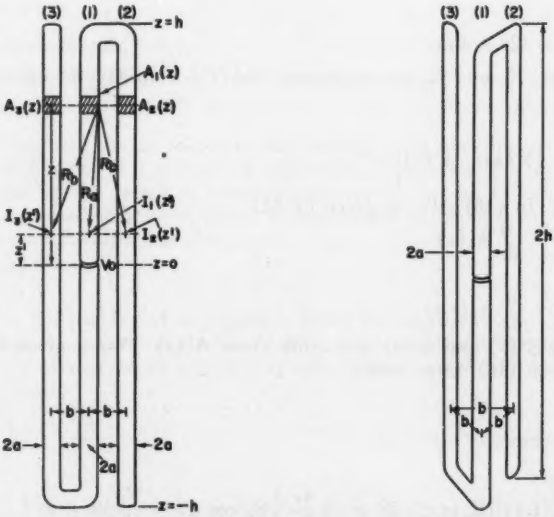


Figure 1A (left). Symmetrically Folded Antenna. The vector potential $A_1(z)$ is due to the currents $I_1(z)$, $I_2(z)$, and $I_3(z)$. The figure is distorted in an attempt to portray correctly the distances R_a and R_b. In any plane perpendicular o the wire axes ($-h<z<h$) the center of the wires of the actual antenna analyzed form the vertices of an equilateral triangle.

Figure 1B (right). Symmetrically Folded Antenna in perspective.

Subject to (22), (8) yields

$$A_1(z) = \frac{-j}{c}\left\{C_1 \cos \beta z + \frac{V_o}{2} \sin \beta/z/\right\} \quad \ldots\ldots (23)$$

$$A_2(z) = \frac{-j}{c}\left\{C_2 \cos \beta z + C_3 \sin \beta z\right\} \quad \ldots\ldots (24)$$

$$A_3(z) = \frac{-j}{c}\left\{C_2 \cos \beta z - C_3 \sin \beta z\right\} \quad \ldots\ldots (25)$$

It is evident that these expressions give the total vector potential at the point z on the surface of the wire designated by the subscript. The vector characterization of A has been dropped. It is understood that A points in the z direction and is functionally dependent on the value of the coordinate z. Henceforth $A_z(z)$ and $I_z(z)$ are written A(z) and I(z), respectively.

The vector potential on the surface of any one of the three conductors is due to the currents $I_1(z)$, $I_2(z)$ and $I_3(z)$ flowing along wires (1), (2) and (3), respectively. Therefore, to obtain a second expression for $A_1(z)$ requires the addition of three integrals like (11); proper account being taken of retardation in the distance factor $R\alpha = \sqrt{(z-z')^2 + a^2}$. $A_2(z)$ and $A_3(z)$ are formulated in the same way.

One is now in a position to write down the three simultaneous integral equations governing the operation of this circuit. They are:

$$\int_{-h}^{h} I_1(z')K_a(z,z')\,dz' + \int_{-h}^{h} I_2(z')K_b(z,z')\,dz' + \int_{-h}^{h} I_3(z')K_b(z,z')\,dz'$$

$$= -j\frac{4\pi}{R_c}\left\{C_1 \cos \beta z + \frac{V_o}{2} \sin B/z/\right\} \quad \ldots\ldots (26)$$

$$\int_{-h}^{h} I_1(z')K_b(z,z')\,dz' + \int_{-h}^{h} I_2(z')K_a(z,z')\,dz' + \int_{-h}^{h} I_3(z')K_b(z,z')\,dz'$$

$$= -j\frac{4\pi}{R_c}\left\{C_2 \cos \beta z + C_3 \sin \beta z)\right\} \quad \ldots\ldots (27)$$

$$\int_{-h}^{h} I_1(z')K_b(z,z')\,dz' + \int_{-h}^{h} I_2(z')K_b(z,z')\,dz' + \int_{-h}^{h} I_3(z')K_a(z,z')\,dz'$$

$$= -j\frac{4\pi}{R_c}\left\{C_2 \cos \beta z - C_3 \sin \beta z\right\} \quad \ldots\ldots (28)$$

$K_a(z,z')$ and $K_b(z,z')$ are defined by (12).

It is now apparent that along the surface of each conductor the solution of the one dimensional wave equation (8), in terms of the total vector potential, is the sum of three integrals involving the unknown currents $I_1(z)$, $I_2(z)$ and $I_3(z)$ under the integral sign.

III. TRANSFORMATION OF THE THREE INTEGRAL EQUATIONS

The first step in the solution of the set (26) to (28) inclusive, is to express the first integral in (26) in the form

$$\int_{-h}^{h} I_1(z')K_a(z,z')\,dz' = \int_{-h}^{h} I_1(z')K_d(z,z')\,dz' + \int_{-h}^{h} I_1(z')\left\{K_a(z,z') - K_d(z,z')\right\}dz' \quad (29)$$

where $K_d(z,z')$ is given by (12) with α replaced by d. In (29) d is an as yet arbitrary small distance that satisfies the condition

$$\beta d << 1 \quad (30)$$

Clearly (29) is an identity. Now consider the second integral on the right hand side of (29). Since βa and βd are both small (see (9) and (30)) this integral has a value different from zero only when $z \approx z'$. In this region the amplitude of the current is slowly varient; hence $I_1(z)$ may be removed from under the sign of integration. When this has been done the limits of integration $-h \leq z' \leq h$ may be replaced by $-\infty \leq z' \leq \infty$. Thus

$$\int_{-h}^{h} I_1(z')\left\{K_a(z,z')\,dz' - K_d(z,z')\right\}dz' = I_1(z)\int_{-\infty}^{\infty}\left\{K_a(z,z') - K_d(z,z')\right\}dz' = I_1(z)\lambda_a \quad (31)$$

Here

$$\lambda_a = 2\ln\frac{d}{a} \quad (32)$$

Equation (29) now becomes

$$\int_{-h}^{h} I_1(z')K_a(z,z')\,dz' = \int_{-h}^{h} I_1(z')K_d(z,z')\,dz' + I_1(z)\lambda a \quad (33)$$

All other integrals are treated in precisely the same way. More generally,

$$\int_{-h}^{h} I_n(z')K_a(z,z)'\,dz' = \int_{-h}^{h} I_n(z')K_d(z,z')dz' + I_n(z)\lambda\alpha \quad (34)$$

When $\alpha = b$, (34) is valid only if

$$\beta b << 1 \quad (35)$$

Application of (34) to the set (26) to (28), inclusive, yields

$$J_d(z) + I_1(z)\lambda_a + I_2(z)\lambda_b + I_3(z)\lambda_b = -j\frac{4\pi}{R_c}\left\{C_1\cos\beta z + \frac{V_o}{2}\sin\beta/z/\right\} \quad (36)$$

$$J_d(z) + I_1(z)\lambda_b + I_2(z)\lambda_a + I_3(z)\lambda_b = -j\frac{4\pi}{R_c}\left\{C_2\cos\beta z + C_3\sin\beta z\right\} \quad (37)$$

$$J_d(z) + I_1(z)\lambda_b + I_2(z)\lambda_b + I_3(z)\lambda_a = -j\frac{4\pi}{R_c}\left\{C_2\cos\beta z - \sin\beta z\right\} \quad (38)$$

The notation

$$J_d(z) = \int_{-h}^{h} \left\{ I_1(z') + I_2(z') + I_3(z') \right\} K_d(z,z')\, dz' = \int_{-h}^{h} I_T(z,z') K_d(z,z')\, dz' \quad (39)$$

is employed. λ_b has the value

$$\lambda_b = 2 \ln \frac{d}{b} \quad (40)$$

IV. REDUCTION OF THE THREE INTEGRAL EQUATIONS TO ONE EQUATION OF KNOWN SOLUTION

Adding equations (36) to (38), inclusive, yields

$$3J_d(z) + \left\{ I_1(z) + I_2(z) + I_3(z) \right\} \left\{ \lambda_a + 2\lambda_b \right\} = -j \frac{4\pi}{R_c} \left\{ (C_1 + 2C_2) \cos \beta z + \frac{V_o}{2} \sin \beta |z| \right\} \quad (41)$$

Since no restriction has been imposed on λ_a or λ_b, one is free to require

$$\lambda_a = -2\lambda_b \quad (42)$$

That is,

$$2 \ln \frac{d}{a} = -4 \ln \frac{d}{b} \quad (43)$$

so that

$$d = \sqrt[3]{ab^2} < b \quad (44)$$

Employing (42), (41) becomes

$$J_d(z) = -j \frac{4\pi}{R_c} \left\{ \frac{C_1 + 2C_2}{3} \cos \beta z + \frac{V_o}{6} \sin \beta |z| \right\} \quad (45)$$

This equation is identically the same as (20) provided one replaces d by a, $I_T(z)$ by $I(z)$, $(C_1 + 2C_2)/3$ by K_1 and $V_o/6$ by $V_K/2$. Now K_1 is determined by (15). If this boundary condition also holds in the present problem, the solution of (45) may be obtained directly from the solution of (20). That the boundary condition on the total current is the same is evident from a consideration of Figure (1). From Kirchhoff's law, $I_1(h) + I_2(h) = 0$. Also $I_3(h) = 0$. Similarly, $I_1(-h) + I_3(-h) = 0$ and $I_2(-h) = 0$. Accordingly,

$$I_T(\pm h) = 0 \quad (46)$$

The solution of (45) is therefore

$$Z_d = \frac{V_o}{3I_T(0)} \quad (47)$$

Here Z_d is the driving point impedance of a symmetrical center-driven antenna of half-length h and radius d. It is of interest to point out that antenna equations formally similar to (45) may be obtained in the case of all center-driven folded wire antennas, irrespective of whether the wires are of the same diameter or not. Hence the tables for Z_d and Y_d given in the Appendix are indispensable to the solution of problems in this field.

V. TRANSMISSION LINE EQUATIONS

The driving point admittance of the symmetrically folded antena being studied is

$$Y_{in} = \frac{I_1(0)}{V_o} \quad (48)$$

Accordingly, it is necessary to obtain the relationships existing between the fictitious current $I_T(0)$ and the currents $I_1(0)$, $I_2(0)$ and $I_3(0)$.

Subtracting (37) from (36), and then (38) from (36),

$$I_1(z) - I_2(z) = -j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1 - C_2)\cos\beta z + \frac{V_o}{2}\sin\beta/z/ - C_3\sin\beta z\right\} \quad \ldots\ (49)$$

$$I_1(z) - I_3(z) = -j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1 - C_2)\cos\beta z + \frac{V_o}{2}\sin\beta/z/ - C_3\sin\beta z\right\} \quad \ldots\ (50)$$

Here

$$\lambda_{ab} = \lambda_a - \lambda_b = 2\left\{\ln\frac{d}{a} - \ln\frac{d}{b}\right\} = 2\ln\frac{b}{a} \quad \ldots\ (51)$$

The expressions (49) and (50) constitute a system of linear equations for the current. They are true transmission line equations, whereas (45) is a true antenna equation. Thus a folded wire circuit operates simultaneously in an antenna and transmission line mode, the impedance depending on a synthesis of antenna and transmission line effects.

VI. CHARGE CONTINUITY

In analyzing folded antennas it is important to insure that all constants of integration are evaluated in such a way that no generators appear in the circuit where there are none. This condition is met if one requires the charge density $q(z)$ to be continuous except where the structure is driven. The equation of continuity of electric charge in this application is

$$\frac{\delta I(z)}{\delta z} = -j\omega q(z) \quad \ldots\ (52)$$

Since no generator exists in the short bar connecting wires (1) and (2) together at $z = h$, (52) yields

$$\left\{\frac{\delta I_1(z)}{\delta z} - \frac{I_2(z)}{\delta z}\right\}_{z=h} = 0 \quad \ldots\ (53)$$

By symmetry the same result is obtained by performing the operation

$$\left\{\frac{\delta I_1(z)}{\delta z} - \frac{\delta I_3(z)}{\delta z}\right\}_{z=-h} = 0 \quad \ldots\ (54)$$

Application of (53) to (49) leads to one expression relating the constants. It is

$$-(C_1 - C_2)\sin\beta h + \frac{V_o}{2}\cos\beta h - C_3\cos\beta h = 0 \quad \ldots\ (55)$$

VII. BOUNDARY CONDITION ON THE CURRENT

If one sets $z = h$ in (50) it follows that

$$I_1(h) = -j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1 - C_2)\cos\beta h + \frac{V_o}{2}\sin\beta h + C_3\sin\beta h\right\} \quad \ldots\ (56)$$

because $I_3(h) = 0$. Also from (49), for $z = h$,

$$I_1(h) - I_2(h) = -j\frac{4\pi}{\lambda_{ab}R_c}\left\{(C_1 - C_2)\cos\beta h + \frac{V_o}{2}\sin\beta h - C_3\sin\beta h\right\} \quad \ldots\ (57)$$

Now the boundary condition on the current at the juncture of wires (1) and (2) is

$$I_1(h) + I_2(h) = 0 \quad \ldots\ldots (58)$$

Subtracting twice (56) from (57) and using (58) results in a second expression relating the constants. It is

$$(C_1 - C_2)\cos\beta h + \frac{V_o}{2}\sin\beta h + 3C_3 \sin\beta h = 0 \quad \ldots\ldots (59)$$

VIII. DRIVING POINT ADMITTANCE

On setting $z = 0$ in (49) and rearranging,

$$I_1(0) = I_2(0) - j\frac{4\pi}{\lambda_{ab}R_c}\left\{C_1 - C_2\right\} \quad \ldots\ldots (60)$$

From 50, for $z = 0$,

$$I_1(0) = I_3(0) - j\frac{4\pi}{\lambda_{ab}R_c}\left\{C_1 - C_2\right\} \quad \ldots\ldots (61)$$

On summing (60) and (61), and then adding $I_1(z)$ to both sides of the resulting expression,

$$I_1(0) = \frac{I_T(0)}{3} - j\frac{8\pi}{3\lambda_{ab}R_c}(C_1 - C_2) \quad \ldots\ldots (62)$$

The simultaneous solution of equations (55) and (59) gives

$$C_1 - C_2 = \frac{V_o \sin 2\beta h}{1 - 2\cos 2\beta h} \quad \ldots\ldots (63)$$

and

$$C_3 = -\frac{V_o}{2(1 - 2\cos 2\beta h)} \quad \ldots\ldots (64)$$

Substituting (63) into (62) and dividing by V_o,

$$Y_{in} = \frac{I_1(0)}{V_o} = \frac{I_T(0)}{3V_o} - j\frac{8\pi}{3\lambda_{ab}R_c}\,\frac{\sin 2\beta h}{(1 - 2\cos 2\beta h)} \quad \ldots\ldots (65)$$

Introducing into (65) the value of $I_T(0)$ from (47), λ_{ab} from (51) and R_c from (14), the final formula for the driving point admittance of the symmetrically folded antenna is obtained. It is

$$Y_{in} = \frac{1}{9Z_d} - j\frac{\sin 2\beta h}{90(1 - 2\cos 2\beta h)\ln\frac{b}{a}} \text{ mhos} \quad \ldots\ldots (66)$$

Here Z_d is the impedance in ohms of a symmetrical center-driven antenna of half-length h and radius $d = \sqrt[3]{ab^2}$. Valid results may be expected from use of this formula provided $a << h$, $\beta a << 1$ and $\beta b << 1$.

IX. CONVENTIONAL THREE WIRE FOLDED DIPOLE

One objective of this paper is to compare the operational characteristics of the symmetrically folded antenna with those of the conventional three wire folded dipole. In the interest of brevity the detailed derivation of the formula

for the input admittance of the conventional folded dipole will not be given here; however the high points in the analysis will be presented to aid the reader in solving the problem, if he so desires.

Figure 2 is an illustration of the circuit. The center-driven wire is designated (1). The other wires are numbered (2) and (3). All conductors are connected together at each end of the structure by means of shorting bars. The cross-section of the antenna-transmission line, taken at right angles to the wire axes, is an equilateral triangle of leg length b.

The electrical properties of the three-element folded dipole may be deducted from the two integral equations

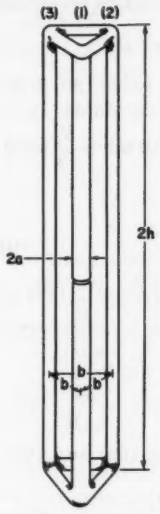


Figure 2. Conventional Three Wire Folded Dipole in perspective.

$$J_d(z) + I_1(z)\lambda_a + 2I_2(z)\lambda_b = -j\frac{4\pi}{R_c}\left\{C_1 \cos\beta z + \frac{V_o}{2}\sin\beta/z/\right\} \quad \ldots\ldots (67)$$

$$J_d(z) + I_1(z)\lambda_b + I_2(z)\left\{\lambda_a + \lambda_b\right\} = -j\frac{4\pi}{R_c}\left\{C_2 \cos\beta z\right\} \quad \ldots\ldots (68)$$

Equation (67) applies to the conductor designated (1). Equation (68) applies to either wire (2) or (3) because from symmetry considerations $A_2(z) = A_3(z)$ and $I_2(z) = I_3(z)$. The integral $J_d(z)$ is defined by (39); however, it is to be observed that in this case $I_T(z) = I_1(z) + 2I_2(z)$.

The effective radius of the structure is given by (44). Charge continuity requires

$$\left\{\frac{\delta I_1(z)}{\delta z} - \frac{\delta I_2(z)}{\delta z}\right\}_{z = \pm h} = 0 \quad \ldots\ldots (69)$$

The boundary condition on the current is given by (46).

The constant $(C_1 - C_2)$ has the value

$$C_1 - C_2 = \frac{V_o}{2}\cot\beta h \quad \ldots\ldots (70)$$

The driving point admittance is

$$Y_{in} = \frac{1}{9Z_d} - j\frac{\cot\beta h}{180 \ln\frac{b}{a}} \text{ mhos} \quad \ldots\ldots (71)$$

Formula (71) is valid only if $a << h$, $\beta a << 1$ and $\beta b << 1$. As before, Z_d is the driving point impedance of a symmetrical center-driven antenna of half-length h and radius $d = \sqrt[3]{ab^2}$.

X. NUMERICAL ANALYSIS

In order to compare the band width, i.e., impedance versus frequency characteristic of a typical symmetrically folded antenna and a conventional three-element folded dipole, the driving point impedances of each have been computed subject to the following conditions:

$\Omega = 10$ or $h/d = 74.207$

$b = 10a$.

Ω is defined by the formula

$$\Omega = 2 \ln \frac{2h}{d}, \quad \ldots\ldots (72)$$

and $d = \sqrt[3]{ab^2}$.

Table A, applying to the symmetrically folded antenna, was computed from (66) using Table 4 (or Table 12) in the Appendix. Table B, applying to the three-element folded dipole, was computed from (71) using the same tables.

Table A

βh	Z_{in} (ohms)	ρ (600 ohm line)
0.5	0.00 − j 19.90	
0.5236	0.00 + j 0.00	
1.1	73.10 + j 839.0	
1.2	744.9 + j 1600	7.66
1.3	1581 − j 291.2	2.74
1.4	688.4 − j 124.6	1.27
1.5	622.0 + j 183.0	1.35
1.5708	778.5 + j 375.3	1.82
1.6	902.0 + j 443.1	2.04
1.7	1792 + j 190.9	3.02
1.8	1310 − j 1309	
1.9	363.0 − j 1059	
2.0	124.5 − j 718.2	
2.2	23.50 − j 366.6	
2.3	10.67 − j 260.6	
2.6	0.0280 − j 14.77	
2.618	0.00 + j 0.00	
2.9	357.8 + j 7.146	3.02
3.1	1747 + j 3572	
3.1416	2250 − j 3924	
3.2	179.1 − j 1257	

Table B

β h	Z_{in} (ohms)	ρ (600 ohm line)
0.5	0.0612 + j 234.7	
1.1	261.8 + j 1572	
1.2	3170 + j 1793	7.0
1.3	1216 − j 713.5	2.87
1.4	657.8 − j 186.4	1.36
1.5	634.4 + j 162.0	1.31
1.5708	778.5 + j 375.3	1.82
1.6	882.7 + j 457.0	2.06
1.7	1662 + j 498.2	3.06
1.8	2142 − j 1011	
1.9	805.3 − j 1461	
2.0	287.0 − j 1069	
2.3	36.37 − j 480.0	
2.6	7.806 − j 247.8	
2.9	1.155 − j 100.9	
3.1416	0.00 + j 0.00	
3.2	0.659 + j 24.37	
3.4	1.496 + j 11.29	

Assume that a transmission line having a characteristic impedance $Z_c = 600 + j\,0$ ohms is used to connect a radio transmitter to either antenna. The standing wave ratio ρ, existing along the line, may be calculated from the formula

$$\rho = \frac{1+/\Gamma/}{1-/\Gamma/} \qquad (73)$$

where

$$/\Gamma/ = \frac{/Z_{in} + Z_c/}{/Z_{in} - Z_c/} \qquad (74)$$

In the range $1.3 \leq \beta\,h \leq 1.7$, $\rho \leq 3.06$ for both antennas. Thus the bandpass characteristics in this range of β h are almost identical. However, when the conventional folded dipole degenerates into a pure transmission line at $\beta\,h = \pi$, i.e., the radiation resistance goes to zero, the symmetrically folded antenna has a non-vanishing radiation resistance. In fact at $\beta\,h = 2.9$, $\rho = 3.02$. But observe that the band width in the vicinity of $\beta\,h = 2.9$ is much narrower than in the vicinity of $\beta\,h = 1.5$. Clearly when $\beta\,h \approx (2n-1)\frac{\pi}{2}$ the folded dipole acts like an antenna. On the other hand the symmetrically folded structure behaves like an antenna when $\beta\,h \approx n\frac{\pi}{2}$. Here n is any positive integer.

Tables A and B furnish useful design information. For example, suppose one desires to construct a symmetrically folded antenna to cover a band of frequencies centered on 12 mcs (25 meters). If $\Omega = 10$ and $b = 10a$ one might select $\beta h = 1.5$. Then h (the antenna half-length) is

$$h = \frac{1.5}{\beta} = \frac{1.5\lambda}{2\pi} = \frac{1.5 \times 25}{6.283} = 5.968 \text{ meters}$$

(or 19.58 feet). Now, having determined h, one may calculate the frequency limits of the antenna for $\rho \leqq 3.02$ on a 600 ohm feed line. These limits are set by $\beta h = 1.3$ and $\beta h = 1.7$. Since

$$\beta h = \frac{2\pi}{\lambda} h = \frac{2\pi f}{c} h, \; c = 3 \times 10^8 \text{ meters/sec. and } h = 5.968 \text{ meters},$$

10.4 mcs $\leqq f \leqq$ 13.6 mcs. Thus the transmitter frequency may be varied over the range 10.4 to 13.6 megacycles per second without the standing wave ratio on the 600 ohm line extending 3. It is conceivable that some further improvement in antenna band width could be effected by the employment of an ancillary matching network.

APPENDIX

Universal Tables for Folded Antennas

Tables of the Function Z_a		*Tables of the Function* Y_a	
Table 1	$\Omega = 7$, $\beta h \leqq 4$	Table 9	$\Omega = 7$, $\beta h \leqq 4$
Table 2	$\Omega = 8$, $\beta h \leqq 4$	Table 10	$\Omega = 8$, $\beta h \leqq 4$
Table 3	$\Omega = 9$, $\beta h \leqq 4$	Table 11	$\Omega = 9$, $\beta h \leqq 4$
Table 4	$\Omega = 10$, $\beta h \leqq 7$	Table 12	$\Omega = 10$, $\beta h \leqq 7$
Table 5	$\Omega = 11$, $\beta h \leqq 4$	Table 13	$\Omega = 11$, $\beta h \leqq 4$
Table 6	$\Omega = 12.5$, $\beta h \leqq 7$	Table 14	$\Omega = 12.5$, $\beta h \leqq 7$
Table 7	$\Omega = 15$, $\beta h \leqq 7$	Table 15	$\Omega = 15$, $\beta h \leqq 7$
Table 8	$\Omega = 20$ $\beta h \leqq 7$	Table 16	$\Omega = 20$, $\beta h \leqq 7$

Table of the Critical Values of the Function Z_a *and* Y_a

Table 17 $\quad \beta h = \frac{\pi}{2}$, $7 \leqq \Omega \leqq 20$

TABLE 1

$\Omega = 7$

βh	Z_a (ohms)
0.5	4.985 −j375.6
0.7	10.38 −j237.4
0.9	18.66 −j150.6
1.1	31.31 −j 85.40
1.2	39.84 −j 56.87
1.3	50.64 −j 30.05
1.4	64.06 −j 4.582
1.5	80.83 +j 19.70
1.6	101.8 +j 42.51
1.7	129.1 +j 63.01
1.8	164.8 +j 78.42
1.9	210.2 +j 83.60
2.0	264.0 +j 69.99
2.1	315.9 +j 27.76
2.2	344.6 −j 44.82
2.3	330.5 −j127.0
2.4	279.0 −j187.6
2.5	215.2 −j214.4
2.6	158.2 −j215.2
2.7	114.8 −j202.5
2.8	83.36 −j184.6
2.9	61.37 −j165.9
3.0	46.08 −j148.1
3.1	35.77 −j132.0
3.2	28.95 −j117.8
3.3	24.72 −j105.0
3.4	22.43 −j 93.61
3.5	21.54 −j 83.35
3.6	21.82 −j 73.79
3.7	22.96 −j 64.61
3.8	24.87 −j 55.50
3.9	27.61 −j 46.21
4.0	31.20 −j 36.40

TABLE 2

$\Omega = 8$

βh	Z_a (ohms)
0.5	4.984 −j484.5
0.7	10.30 −j308.4
0.9	18.49 −j198.2
1.1	30.70 −j116.0
1.2	38.94 −j 80.11
1.3	49.14 −j 46.43
1.4	61.83 −j 14.11
1.5	77.59 +j 17.28
1.6	97.41 +j 47.93
1.7	123.0 +j 77.81
1.8	157.1 +j105.6
1.9	202.1 +j128.5
2.0	261.1 +j140.7
2.1	334.1 +j131.5
2.2	411.6 +j 86.47
2.3	439.4 −j 2.858
2.4	474.3 −j118.5
2.5	421.2 −j218.4
2.6	338.9 −j274.3
2.7	259.0 −j290.3
2.8	194.5 −j282.1
2.9	146.7 −j263.1
3.0	112.6 −j240.6
3.1	88.63 −j218.0
3.2	72.01 −j196.8
3.3	60.59 −j177.4
3.4	52.92 −j159.8
3.5	47.88 −j143.7
3.6	44.92 −j128.7
3.7	43.34 −j114.5
3.8	42.96 −j100.5
3.9	43.61 −j 86.53
4.0	45.33 −j 72.25

TABLE 3

$\Omega = 9$

βh	Z_d (ohms)
0.5	4.988 −j594.6
0.7	10.31 −j379.8
0.9	18.39 −j245.9
1.1	30.31 −j146.7
1.2	38.29 −j103.6
1.3	48.15 −j 63.05
1.4	60.27 −j 24.07
1.5	75.34 +j 14.00
1.6	94.16 +j 51.69
1.7	118.5 +j 89.54
1.8	150.3 +j126.9
1.9	193.1 +j162.9
2.0	250.6 +j194.0
2.1	327.3 +j213.4
2.2	425.4 +j207.1
2.3	534.9 +j154.7
2.4	624.1 +j 40.64
2.5	646.6 −j117.4
2.6	586.5 −j261.8
2.7	480.0 −j349.1
2.8	371.8 −j379.8
2.9	283.1 −j374.5
3.0	217.0 −j352.7
3.1	169.4 −j324.7
3.2	135.4 −j295.8
3.3	111.3 −j268.1
3.4	94.12 −j242.4
3.5	81.77 −j218.6
3.6	73.05 −j196.4
3.7	66.77 −j175.3
3.8	62.46 −j154.9
3.9	59.86 −j134.8
4.0	58.68 −j114.5

TABLE 5

$\Omega = 11$

βh	Z_d (ohms)
0.5	4.977 −j815.1
0.7	10.30 −j522.9
0.9	18.24 −j341.7
1.1	29.81 −j208.3
1.2	37.51 −j150.5
1.3	46.87 −j 96.47
1.4	58.31 −j 44.56
1.5	72.38 +j 6.378
1.6	89.92 +j 57.21
1.7	112.0 +j109.0
1.8	141.1 +j162.3
1.9	179.2 +j217.6
2.0	231.0 +j274.6
2.1	302.2 +j331.2
2.2	400.7 +j382.2
2.3	537.6 +j413.7
2.4	715.9 +j398.1
2.5	917.1 +j292.0
2.6	1070. +j 64.78
2.7	1080. −j230.7
2.8	939.9 −j471.5
2.9	740.9 −j592.2
3.0	560.2 −j618.5
3.1	421.6 −j594.4
3.2	321.5 −j549.8
3.3	250.6 −j500.0
3.4	199.9 −j451.3
3.5	163.0 −j405.1
3.6	135.7 −j361.4
3.7	115.5 −j321.0
3.8	100.2 −j282.7
3.9	88.98 −j245.9
4.0	81.06 −j209.8

TABLE 4

$\Omega = 10$

βh	Z_d (ohms)	βh	Z_d (ohms)
0.5	4.988 − j704.8	4.0	70.61 − j160.8
0.7	10.30 − j451.3	4.1	68.42 − j133.3
0.9	18.29 − j293.8	4.2	68.61 − j105.4
		4.3	71.37 − j 76.93
1.1	30.02 − j177.4	4.4	77.17 − j 47.77
1.2	37.84 − j127.1		
1.3	47.41 − j 79.76	4.5	86.62 − j 17.91
1.4	59.15 − j 34.27	4.6	100.6 + j 12.49
		4.7	120.2 + j 43.02
1.5	73.65 + j 10.30	4.8	147.0 + j 72.76
1.6	91.73 + j 54.72	4.9	182.8 + j 99.87
1.7	114.8 + j 99.67		
1.8	145.2 + j145.5	5.0	229.5 + j121.0
1.9	185.5 + j191.8	5.1	287.9 + j130.5
		5.2	356.3 + j120.5
2.0	240.2 + j237.1	5.3	425.9 + j 82.34
2.1	314.8 + j277.6	5.4	480.6 + j 12.16
2.2	416.1 + j303.8		
2.3	547.5 + j296.8	5.5	502.0 − j 79.40
2.4	697.1 + j227.3	5.6	483.1 − j170.1
		5.7	433.4 − j240.3
2.5	820.4 + j 71.37	5.8	370.0 − j282.3
2.6	849.9 − j145.6	5.9	307.5 − j299.7
2.7	765.5 − j339.9		
2.8	622.2 − j452.6	6.0	253.0 − j300.1
2.9	479.4 − j489.4	6.1	208.4 − j290.3
		6.2	173.2 − j275.2
3.0	364.5 − j480.4	6.3	145.8 − j257.4
3.1	280.0 − j450.6	6.4	124.7 − j238.8
3.2	219.6 − j414.0		
3.3	175.6 − j376.2	6.5	108.8 − j220.4
3.4	143.8 − j339.9	6.6	96.66 − j202.2
		6.7	87.46 − j184.4
3.5	120.9 − j306.1	6.8	80.60 − j167.0
3.6	103.9 − j274.5	6.9	75.60 − j149.6
3.7	91.22 − j244.7	7.0	72.19 − j132.0
3.8	81.84 − j216.4		
3.9	75.09 − j188.4		

TABLE 6

$\Omega = 12.5$

βh	Z_d (ohms)	βh	Z_d (ohms)
0.5	5.005 −j980.6	3.6	182.0 −j504.4
0.7	10.29 −j630.3	3.7	149.0 −j444.2
0.9	18.18 −j413.6	3.8	124.6 −j388.3
		3.9	106.7 −j336.0
1.1	29.60 −j254.2	4.0	93.91 −j286.1
1.2	37.13 −j185.7		
1.3	46.27 −j121.6	4.1	85.39 −j237.9
1.4	57.39 −j 60.07	4.2	80.70 −j191.0
1.5	71.02 +j 0.2681	4.3	79.63 −j144.7
		4.4	82.23 −j 98.56
1.6	87.90 +j 60.62	4.5	88.77 −j 52.14
1.7	109.2 +j122.2		
1.8	136.6 +j186.1	4.6	99.77 −j 5.109
1.9	172.7 +j253.4	4.7	116.0 +j 42.88
2.0	221.2 +j325.0	4.8	138.8 +j 91.94
		4.9	169.9 +j141.9
2.1	287.7 +j400.9	5.0	211.8 +j192.2
2.2	381.2 +j479.5		
2.3	514.5 +j554.2	5.1	268.1 +j241.1
2.4	705.0 +j606.5	5.2	342.9 +j284.5
2.5	965.3 +j594.0	5.3	440.5 +j315.0
		5.4	562.7 +j319.5
2.6	1268. +j443.0	5.5	703.2 +j278.7
2.7	1499. +j 93.51		
2.8	1490. −j358.7	5.6	838.4 +j174.0
2.9	1248. −j696.0	5.7	927.2 +j 5.512
3.0	943.6 −j835.5	5.8	933.5 −j191.0
		5.9	856.6 −j361.6
3.1	689.7 −j841.4	6.0	734.5 −j472.9
3.2	506.0 −j787.8		
3.3	378.4 −j714.7	6.1	605.1 −j525.0
3.4	289.7 −j640.3	6.2	490.3 −j534.5
3.5	227.1 −j569.4	6.3	396.7 −j519.1
		6.4	323.1 −j490.5
		6.5	266.3 −j455.9
		6.6	222.1 −j419.3
		6.7	187.7 −j382.2
		6.8	160.7 −j345.5
		6.9	139.6 −j309.5
		7.0	123.1 −j273.7

TABLE 7

$\Omega = 15$

βh	Z_d (ohms)	βh	Z_d (ohms)
0.5	5.000 −j1256.	4.0	110.0 −j 417.0
0.7	10.28 −j 809.3	4.1	97.25 −j 345.9
0.9	18.13 −j 533.2	4.2	89.70 −j 278.2
		4.3	85.53 −j 212.5
1.1	29.36 −j 330.9	4.4	85.48 −j 148.5
1.2	36.73 −j 244.4		
1.3	45.62 −j 163.5	4.5	89.91 −j 85.51
1.4	56.38 −j 86.00	4.6	99.14 −j 22.67
		4.7	113.5 +j 41.25
1.5	69.46 −j 10.23	4.8	133.9 +j 107.2
1.6	85.53 +j 65.50	4.9	161.9 +j 175.7
1.7	105.7 +j 142.8		
1.8	131.5 +j 223.4	5.0	199.8 +j 247.1
1.9	165.0 +j 309.1	5.1	250.9 +j 321.7
		5.2	320.2 +j 399.2
2.0	209.4 +j 401.7	5.3	414.5 +j 476.5
2.1	269.8 +j 503.3	5.4	543.5 +j 547.5
2.2	354.8 +j 616.3		
2.3	476.3 +j 740.3	5.5	717.7 +j 597.1
2.4	656.3 +j 871.2	5.6	944.4 +j 596.8
		5.7	1210. +j 500.0
2.5	929.4 +j 988.5	5.8	1452. +j 262.2
2.6	1330. +j1045.	5.9	1564. −j 99.34
2.7	1878. +j 860.4		
2.8	2361. +j 278.9	6.0	1481. −j 469.4
2.9	2362. −j 570.8	6.1	1258. −j 726.5
		6.2	1001. −j 843.2
3.0	1870. −j1159.	6.3	775.0 −j 862.1
3.1	1312. −j1328.	6.4	598.4 −j 826.7
3.2	898.8 −j1273.		
3.3	626.4 −j1146.	6.5	465.6 −j 766.8
3.4	449.8 −j1008.	6.6	367.4 −j 698.8
		6.7	293.8 −j 629.8
3.5	332.8 −j 881.1	6.8	238.5 −j 562.5
3.6	253.2 −j 767.3	6.9	196.6 −j 497.9
3.7	197.6 −j 666.1	7.0	164.7 −j 436.0
3.8	158.1 −j 574.7		
3.9	129.9 −j 492.9		

TABLE 8

$\Omega = 20$

βh	Z_a (ohms)
0.5	5.030 −j1809.
0.9	18.93 −j 885.4
1.3	44.92 −j 247.3
1.4	55.22 −j 138.0
1.5	67.65 −j 31.59
1.6	82.98 +j 74.58
1.7	101.9 +j 182.9
1.8	125.3 +j 295.1
1.9	155.5 +j 414.3
2.0	195.3 +j 544.3
2.1	247.7 +j 688.9
2.2	320.0 +j 853.0
2.3	432.1 +j1043.
2.4	576.0 +j1269.
2.5	815.1 +j1538.
2.6	1210. +j1850.
2.7	1895. +j2150.
2.8	3075. +j2165.
2.9	4567. +j1104.
3.0	4790. −j1191.
3.1	3262. −j2521.
3.2	1944. −j2570.
3.3	1182. −j2240.
3.4	759.3 −j1890.
3.5	514.3 −j1591.
3.6	364.1 −j1343.
3.7	267.5 −j1138.
3.8	203.1 −j 964.9
3.9	159.4 −j 814.9
4.0	130.4 −j 682.2
4.1	109.3 −j 562.4
4.2	96.62 −j 452.2
4.3	89.81 −j 348.6
4.4	88.04 −j 249.7
4.5	90.90 −j 153.5
4.6	98.33 −j 58.47
4.7	110.6 +j 37.10
4.8	128.5 +j 134.8
4.9	153.1 +j 236.2
5.0	186.2 +j 343.4
5.1	230.8 +j 458.5
5.2	291.1 +j 583.8
5.3	373.8 +j 721.8
5.4	489.7 +j 875.0
5.5	656.1 +j1043.
5.6	901.4 +j1220.
5.7	1270. +j1379.
5.8	1815. +j1442.
5.9	2542. +j1222.
6.0	3215. +j 447.8
6.1	3273. −j 651.4
6.2	2663. −j1497.
6.3	1904. −j1786.
6.4	1314. −j1744.
6.5	916.4 −j1582.
6.6	656.2 −j1395.
6.7	482.1 −j1217.
6.8	363.2 −j1056.
6.9	280.2 −j 912.7
7.0	221.3 −j 786.3

TABLE 9

$\Omega = 7$

βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.035 +j 2.662
0.7	0.184 +j 4.204
0.9	0.810 +j 6.539
1.1	3.785 +j10.32
1.2	8.264 +j11.80
1.3	14.61 +j 8.667
1.4	15.53 +j 1.111
1.5	11.68 −j 2.846
1.6	8.365 −j 3.493
1.7	6.255 −j 3.053
1.8	4.947 −j 2.354
1.9	4.108 −j 1.634
2.0	3.539 −j 0.938
2.1	3.140 −j 0.276
2.2	2.853 +j 0.371
2.3	2.636 +j 1.013
2.4	2.469 +j 1.660
2.5	2.332 +j 2.323
2.6	2.218 +j 3.016
2.7	2.119 +j 3.738
2.8	2.032 +j 4.499
2.9	1.961 +j 5.302
3.0	1.915 +j 6.155
3.1	1.913 +j 7.059
3.2	1.968 +j 8.008
3.3	2.124 +j 9.021
3.4	2.421 +j10.10
3.5	2.906 +j11.25
3.6	3.685 +j12.46
3.7	4.883 +j13.74
3.8	6.723 +j15.00
3.9	9.527 +j15.94
4.0	13.58 +j15.84

TABLE 10

$\Omega = 8$

βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.021 +j 2.063
0.7	0.108 +j 3.239
0.9	0.467 +j 5.002
1.1	2.132 +j 8.056
1.2	4.908 +j10.10
1.3	10.75 +j10.16
1.4	15.37 +j 3.508
1.5	12.28 −j 2.735
1.6	8.262 −j 4.065
1.7	5.807 −j 3.674
1.8	4.384 −j 2.947
1.9	3.523 −j 2.240
2.0	2.968 −j 1.599
2.1	2.592 −j 1.020
2.2	2.327 −j .489
2.3	2.131 +j .013
2.4	1.984 +j .496
2.5	1.871 +j .970
2.6	1.783 +j 1.443
2.7	1.711 +j 1.917
2.8	1.657 +j 2.403
2.9	1.617 +j 2.899
3.0	1.595 +j 3.409
3.1	1.600 +j 3.936
3.2	1.640 +j 4.481
3.3	1.724 +j 5.048
3.4	1.867 +j 5.639
3.5	2.087 +j 6.264
3.6	2.418 +j 6.927
3.7	2.891 +j 7.638
3.8	3.595 +j 8.410
3.9	4.645 +j 9.216
4.0	6.231 +j 9.931

TABLE 11

$\Omega = 9$

βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.014 +j 1.681
0.7	0.071 +j 2.632
0.9	0.302 +j 4.044
1.1	1.351 +j 6.537
1.2	3.138 +j 8.492
1.3	7.650 +j10.02
1.4	14.31 +j 5.715
1.5	12.83 −j 2.384
1.6	8.159 −j 4.479
1.7	5.372 −j 4.059
1.8	3.885 −j 3.280
1.9	3.026 −j 2.552
2.0	2.496 −j 1.932
2.1	2.143 −j 1.397
2.2	1.900 −j 0.925
2.3	1.725 −j 0.499
2.4	1.596 −j 0.104
2.5	1.497 +j 0.272
2.6	1.422 +j 0.635
2.7	1.362 +j 0.991
2.8	1.316 +j 1.344
2.9	1.284 +j 1.699
3.0	1.265 +j 2.056
3.1	1.263 +j 2.421
3.2	1.280 +j 2.796
3.3	1.321 +j 3.182
3.4	1.392 +j 3.585
3.5	1.501 +j 4.013
3.6	1.664 +j 4.473
3.7	1.897 +j 4.981
3.8	2.239 +j 5.554
3.9	2.752 +j 6.198
4.0	3.546 +j 6.918

TABLE 13

$\Omega = 11$

βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.007 +j 1.227
0.7	0.038 +j 1.912
0.9	0.156 +j 2.918
1.1	0.673 +j 4.704
1.2	1.559 +j 6.255
1.3	4.076 +j 8.389
1.4	10.83 +j 8.273
1.5	13.71 −j 1.208
1.6	7.915 −j 5.036
1.7	4.586 −j 4.463
1.8	3.051 −j 3.509
1.9	2.255 −j 2.738
2.0	1.793 −j 2.132
2.1	1.503 −j 1.648
2.2	1.307 −j 1.246
2.3	1.168 −j 0.899
2.4	1.067 −j 0.593
2.5	0.990 −j 0.315
2.6	0.931 −j 0.056
2.7	0.885 +j 0.189
2.8	0.850 +j 0.426
2.9	0.824 +j 0.658
3.0	0.804 +j 0.888
3.1	0.794 +j 1.119
3.2	0.793 +j 1.356
3.3	0.801 +j 1.598
3.4	0.821 +j 1.853
3.5	0.855 +j 2.124
3.6	0.911 +j 2.425
3.7	0.992 +j 2.758
3.8	1.114 +j 3.142
3.9	1.301 +j 3.596
4.0	1.602 +j 4.147

TABLE 12

$\Omega = 10$

βh	Y_d (mhos $\times 10^{-3}$)	βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.0100 +j 1.419	4.0	2.289 +j 5.213
0.7	0.0506 +j 2.214	4.1	3.049 +j 5.938
0.9	0.2110 +j 3.390	4.2	4.337 +j 6.664
		4.3	6.481 +j 6.986
1.1	0.9277 +j 5.481	4.4	9.368 +j 5.800
1.2	2.153 +j 7.229		
1.3	5.507 +j 9.264	4.5	11.07 +j 2.289
1.4	12.66 +j 7.333	4.6	9.791 −j 1.216
		4.7	7.376 −j 2.641
1.5	13.32 −j 1.862	4.8	5.464 −j 2.705
1.6	8.040 −j 4.797	4.9	4.213 −j 2.302
1.7	4.967 −j 4.310		
1.8	3.437 −j 3.444	5.0	3.410 −j 1.798
1.9	2.606 −j 2.693	5.1	2.882 −j 1.306
		5.2	2.519 −j 0.8516
2.0	2.109 −j 2.082	5.3	2.263 −j 0.4376
2.1	1.787 −j 1.576	5.4	2.079 −j 0.0526
2.2	1.568 −j 1.145		
2.3	1.412 −j 0.7652	5.5	1.944 +j 0.3074
2.4	1.297 −j 0.4228	5.6	1.842 +j 0.6486
		5.7	1.765 +j 0.9786
2.5	1.210 −j 0.1052	5.8	1.708 +j 1.303
2.6	1.143 +j 0.1958	5.9	1.668 +j 1.626
2.7	1.091 +j 0.4846		
2.8	1.051 +j 0.7646	6.0	1.642 +j 1.948
2.9	1.021 +j 1.043	6.1	1.632 +j 2.273
		6.2	1.639 +j 2.602
3.0	1.002 +j 1.321	6.3	1.666 +j 2.941
3.1	0.9950 +j 1.601	6.4	1.719 +j 3.290
3.2	0.9999 +j 1.885		
3.3	1.019 +j 2.182	6.5	1.802 +j 3.648
3.4	1.056 +j 2.495	6.6	1.924 +j 4.026
		6.7	2.099 +j 4.427
3.5	1.116 +j 2.826	6.8	2.345 +j 4.857
3.6	1.207 +j 3.186	6.9	2.690 +j 5.324
3.7	1.337 +j 3.588	7.0	3.188 +j 5.830
3.8	1.530 +j 4.044		
3.9	1.825 +j 4.580		

TABLE 14

$\Omega = 12.5$

βh	Y_d (mhos $\times 10^{-3}$)	βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.0052 + j 1.020	3.6	0.6331 + j 1.754
0.7	0.0259 + j 1.586	3.7	0.6787 + j 2.023
0.9	0.1061 + j 2.413	3.8	0.7493 + j 2.335
		3.9	0.8592 + j 2.704
1.1	0.4520 + j 3.881	4.0	1.036 + j 3.155
1.2	1.049 + j 5.211		
1.3	2.733 + j 7.183	4.1	1.336 + j 3.723
1.4	8.314 + j 8.703	4.2	1.877 + j 4.443
1.5	14.08 − j 0.0532	4.3	2.919 + j 5.304
		4.4	4.993 + j 5.982
1.6	7.718 − j 5.322	4.5	8.375 + j 4.919
1.7	4.068 − j 4.552		
1.8	2.563 − j 3.492	4.6	9.997 + j 0.5119
1.9	1.837 − j 2.694	4.7	7.584 − j 2.802
2.0	1.435 − j 2.101	4.8	5.006 − j 3.316
		4.9	3.466 − j 2.895
2.1	1.182 − j 1.646	5.0	2.589 − j 2.348
2.2	1.016 − j 1.278		
2.3	0.8997 − j 0.9691	5.1	2.062 − j 1.854
2.4	0.8150 − j 0.7011	5.2	1.727 − j 1.433
2.5	0.7514 − j 0.4624	5.3	1.502 − j 1.074
		5.4	1.344 − j 0.7629
2.6	0.7025 − j 0.2449	5.5	1.229 − j 0.4871
2.7	0.6643 − j 0.0414		
2.8	0.6343 + j 0.1527	5.6	1.143 − j 0.2373
2.9	0.6111 + j 0.3408	5.7	1.078 − j 0.0064
3.0	0.5941 + j 0.5260	5.8	1.029 + j 0.2107
		5.9	0.9908 + j 0.4183
3.1	0.5827 + j 0.7108	6.0	0.9625 + j 0.6197
3.2	0.5772 + j 0.8986		
3.3	0.5785 + j 1.093	6.1	0.9429 + j 0.8180
3.4	0.5870 + j 1.297	6.2	0.9317 + j 1.016
3.5	0.6042 + j 1.515	6.3	0.9293 + j 1.216
		6.4	0.9367 + j 1.422
		6.5	0.9554 + j 1.636
		6.6	0.9869 + j 1.862
		6.7	1.035 + j 2.108
		6.8	1.107 + j 2.379
		6.9	1.211 + j 2.685
		7.0	1.367 + j 3.039

TABLE 15

$\Omega = 15$

βh	Y_d (mhos $\times 10^{-3}$)	βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.0032 + j 0.7960	4.0	0.5913 + j 2.242
0.7	0.0157 + j 1.236	4.1	0.7380 + j 2.646
0.9	0.0637 + j 1.873	4.2	1.036 + j 3.267
		4.3	1.630 + j 4.049
1.1	0.2660 + j 2.998	4.4	2.881 + j 5.056
1.2	0.6014 + j 4.001		
1.3	1.584 + j 5.676	4.5	5.837 + j 5.509
1.4	5.331 + j 8.132	4.6	9.537 + j 2.218
		4.7	7.784 − j 2.830
1.5	14.09 + j 2.076	4.8	4.551 − j 3.644
1.6	7.369 − j 5.644	4.9	2.840 − j 3.079
1.7	3.348 − j 4.525		
1.8	1.956 − j 3.324	5.0	1.979 − j 2.447
1.9	1.344 − j 2.518	5.1	1.507 − j 1.933
		5.2	1.223 − j 1.524
2.0	1.021 − j 1.957	5.3	1.037 − j 1.195
2.1	0.8273 − j 1.543	5.4	0.9132 − j 0.9199
2.2	0.7016 − j 1.219		
2.3	0.6146 − j 0.9553	5.5	0.8234 − j 0.6850
2.4	0.5518 − j 0.7337	5.6	0.7566 − j 0.4782
		5.7	0.7060 − j 0.2919
2.5	0.5047 − j 0.5397	5.8	0.6672 − j 0.1205
2.6	0.4684 − j 0.3639	5.9	0.6368 + j 0.0404
2.7	0.4401 − j 0.2017		
2.8	0.4177 − j 0.0493	6.0	0.6135 + j 0.1944
2.9	0.4000 + j 0.0967	6.1	0.5961 + j 0.3443
		6.2	0.5839 + j 0.4925
3.0	0.3864 + j 0.2393	6.3	0.5767 + j 0.6415
3.1	0.3765 + j 0.3809	6.4	0.5746 + j 0.7937
3.2	0.3702 + j 0.5242		
3.3	0.3674 + j 0.6720	6.5	0.5784 + j 0.9524
3.4	0.3689 + j 0.8271	6.6	0.5893 + j 1.121
		6.7	0.6083 + j 1.304
3.5	0.3752 + j 0.9932	6.8	0.6389 + j 1.507
3.6	0.3878 + j 1.175	6.9	0.6860 + j 1.738
3.7	0.4093 + j 1.380	7.0	0.7584 + j 2.007
3.8	0.4450 + j 1.616		
3.9	0.4998 + j 1.897		

TABLE 16

$\Omega = 20$

βh	Y_d (mhos $\times 10^{-3}$)	βh	Y_d (mhos $\times 10^{-3}$)
0.5	0.0015 +j 0.5527	4.5	2.846 +j 4.817
0.9	0.0241 +j 1.129	4.6	7.512 +j 4.467
1.3	0.7110 +j 3.914	4.7	8.127 −j 2.726
1.4	2.499 +j 6.246	4.8	3.706 −j 3.887
		4.9	1.932 −j 2.981
1.5	12.14 +j 5.667		
1.6	6.666 −j 5.991	5.0	1.220 −j 2.250
1.7	2.325 −j 4.172	5.1	0.8761 −j 1.740
1.8	1.219 −j 2.871	5.2	0.6841 −j 1.372
1.9	0.7941 −j 2.116	5.3	0.5657 −j 1.092
		5.4	0.4871 −j 0.8703
2.0	0.5840 −j 1.628		
2.1	0.4537 −j 1.285	5.5	0.4320 −j 0.6868
2.2	0.3855 −j 1.028	5.6	0.3917 −j 0.5302
2.3	0.3338 −j 0.8231	5.7	0.3613 −j 0.3925
2.4	0.2966 −j 0.6543	5.8	0.3378 −j 0.2683
		5.9	0.3196 −j 0.1536
2.5	0.2689 −j 0.5075		
2.6	0.2475 −j 0.3786	6.0	0.3053 −j 0.0455
2.7	0.2307 −j 0.2617	6.1	0.2940 +j 0.0585
2.8	0.2174 −j 0.1531	6.2	0.2854 +j 0.1604
2.9	0.2068 −j 0.0500	6.3	0.2793 +j 0.2620
		6.4	0.2755 +j 0.3658
3.0	0.1984 +j 0.0498		
3.1	0.1919 +j 0.1483	6.5	0.2743 +j 0.4734
3.2	0.1872 +j 0.2475	6.6	0.2760 +j 0.5869
3.3	0.1842 +j 0.3493	6.7	0.2811 +j 0.7100
3.4	0.1831 +j 0.4557	6.8	0.2910 +j 0.8465
		6.9	0.3074 +j 1.001
3.5	0.1892 +j 0.5693	7.0	0.3316 +j 1.178
3.6	0.1880 +j 0.6935		
3.7	0.1956 +j 0.8325		
3.8	0.2089 +j 0.9924		
3.9	0.2311 +j 1.182		
4.0	0.2707 +j 1.414		
4.1	0.3329 +j 1.713		
4.2	0.4519 +j 2.115		
4.3	0.6929 +j 2.690		
4.4	1.256 +j 3.562		

TABLE 17

$$\beta h = \frac{\pi}{2}$$

Ω	Z_d (ohms)		Y_d (mhos $\times 10^{-3}$)	
7	95.68	+j 35.85	9.165	−j 3.435
8	91.62	+j 38.98	9.242	−j 3.932
9	88.66	+j 40.68	9.317	−j 4.275
10	86.50	+j 41.70	9.381	−j 4.522
11	84.80	+j 42.36	9.438	−j 4.715
12.5	83.00	+j 43.00	9.499	−j 4.921
15	80.80	+j 43.40	9.604	−j 5.159
20	78.50	+j 43.60	9.736	−j 5.407

THE ECONOMIC SPEED OF TOW ON INLAND WATERWAYS

M. DIMITRYEVIC

The term ECONOMIC *specifies the speed at which the total cost per unit of useful work done is a minimum.*

THE AUTHOR

Mechanical engineer. Member of Yugoslav Society of Engineers and Technicians. Vice-president of Society of Transport Engineers and Technicians. For thirty years leading officer in Technical Department of Inland Waterways Shipping of Yugoslavia as manager of a shipyard, chief engineer and chief planner of transport. At present Head of Shipbuilding Department—Engineering School of Belgrade. Member of editorial staff of the Scientific and Engineering Bulletin of Documentation Center of Yugoslavia.

INTRODUCTORY

The bulk-transport of cargo on inland waterways is very efficiently practiced by means of a convoy, that is, a combination of vessels consisting of a number of dumb barges in tow of a highly powered tug. An alternative arrangement in which the barges are pushed by a "pusher-tug" has been developed in the USA to a high level of efficiency. The comparative advantages of either of these convoys will be considered later; they are both the natural outcome in the development of a cargo-boat intended for traffic in shallow waters, the heavy machinery of the boat having been transferred into a separate hull to provide the necessary freight space in a vessel of small draft.

Though this division of the original powered cargo-ship into a vessel containing the prime mover alone and one or more hulls housing the tonnage could not, obviously, lower the transportation expenses, it did, nevertheless, contribute very effectively to the reduction of freight-rates in general, because it made the bulk-transport on inland waterways possible.

A specific feature of the traffic soon proved the multiplication of hulls to be economic for another reason. The full potentialities of the convoy may only be realized when it is recalled that the average distance between the loading and discharging terminals is relatively very short in traffic on in-

land waterways as compared with the seagoing practice, the necessity of handling cargo in river ports occurring generally rather often. In such circumstances the stoppages also are only too frequent. Now it is currently maintained, and with good reason, that the total cost of the present-day cargo-boat during stopping time is not very much lower than during worktime, which makes her stopping time a heavy loss. As a matter of fact, the transport expenses are ever a loss to the economy of a country; unfortunately they are unavoidable, but at least comprehensible, whereas the stopping time expenses are meaningless, but to a large extent avoidable. They are, indeed, considerably reduced by the previously mentioned separation of the inherently expensive power component (the tug) from the rather low-price freight-space component (the barge) because the latter only needs to stop in ports under cargo-handling equipment, the tug remaining free and available for the next job. Remembering that the participation of the tug in the total cost of transport by means of a convoy generally exceeds 60 percent, the savings realized through her not being affected by the otherwise unavoidable delays in ports may readily be appreciated. In a particular type of convoy, the so-called "integrated tow"—an ingenious arrangement of barges pushed by a tug—so low have sometimes been the transport costs for the conveyance of petroleum products, for instance, that the system could successfully compete even with the pipe-line dispatch.[1]

In addition to this, the separation of the prime mover from the tonnage-space has extended the useful running time of the equivalent cargo-boat employed in irregular, "jobbing," traffic by at least 40 percent.

As regards the formation of convoys it may be stated that, in terms of resistance which is one of the main forces playing a considerable role in this problem of traction, the practice prevailing in USA is hydraulically much more up-to-date than the general European methods, especially in the case of integrated tow. Here are combined the good points of a convoy, i.e., the possibility of transporting large quantities of cargo in shallow draft vessels without its disadvantages of increased resistance owing to multiplication of hulls. By arranging the uniformly shaped barges in a single row to form "the parallel middle body" of a ship and adding a pusher tug immediately behind the last barge, and eventually a shaped body in front of the first one to assist in the steering, a single, long vessel with a very low resistance per unit load may be obtained; this particular type of convoy would be a sensible embodiment of the present-day theory of ship's resistance.

In some cases the specific resistance of some similar built-up cargo-boats has been reduced to 60 percent of the normal convoy formed according to European practices,—a conveniently low figure resulting in the reduction of expenses per ton/mile of about 20 percent.

Reviewing further the hydraulic aspects of convoys other than the integrated tow, the type with barges set in front of the tug is more efficient than the general European counterpart because it allows the wake energy of the front barges to be effectively, though partially, recovered by the rear vessels, thus facilitating the propulsion by way of diminished total resistance. But the European type of convoy, on the other hand, seems to be more suitable for navigation on unregulated waterways, possibly also for the "jobbing" sort of transport of bulk with irregular flow both as regards the time and the direction, and for diverse cargo combined with frequent stoppages en

route, as it is more flexible, offering greater freedom of movement to individual barges. As regards the total resistance, however, its change may be either positive or negative for the propulsion, depending generally on the length of tow-line, the distance of the barges following one another and the way in which they are filed.

But whatever type of convoy be preferred the chief aim in arranging its components after securing the conditions of safe navigation is to produce a complex of minimum resistance.

Both the tug and the barge as former constituent parts of a single vessel have been affected by the separation, but in quite different ways. The tug, for instance, acquired a distinct preference for the tow of large quantities of freight contained in numerous barges, at a very low speed, whereas the barge would work comfortably if loaded with a miniature charge moving at an elevated speed; the latter, of course, regardless of how this would react upon the power developed by the engine. From these contradictory trends of the tug and the barge arose the problem of how to select the optimal speed—optimal meaning economic—and also the economic load of the tow, to suit both these agents in their joint performance of producing useful work. The problem of economic speed was then further complicated by the current, almost invariably present in the inland waterways, because it has made it dependent on its own velocity, in addition to its being already dependent on the manner in which the convoy is formed, and on many other factors, non-mechanical in origin, such as the type, shape and size of the tug and of the barges respectively.

Since by definition the economic speed is a component of *the useful work done*, its magnitude can be determined only when we are in possession of complete data about the force producing the said work. This force is, however, the resultant of several forces and so we are obliged to analyze all the forces acting upon the convoy in motion.

FORCES ACTING UPON THE CONVOY

Subject to analysis will be the convoy consisting of a number—N of barges of similar size and shape, with individual resistance—R_b, in tow of a tug developing a propulsive force—F_p, and having a resistance—R_t. The propulsive force is actually the motive force of the convoy; it is generated by the engine in the tug and delivered on the tip of propeller blades or paddle-wheel floats. When the convoy is in motion this force is partially spent in overcoming the resistance of the tug R_t, the rest appearing as the pull—F_t in the tow-line, i.e., as the tractive force on haulage pintle.

The forces of gravity, buoyancy and inertia are not taken into account because they do not affect our problem in any way.

When the convoy is moving with uniform velocity, all the forces acting on it are in dynamic equilibrium. Thus—

$$F_p = R_t + F_t \text{ and } F_t = R_b \times N \therefore F_p = R_t + R_b \times N \quad . \quad . \quad . \quad . \quad . \quad (1)$$

The same equation (1) holds good for any type of convoy but we have chosen one with a tow-line because the play of various forces, in particular of the tractive force F_t, will be, we presume, easier to follow.

Now let us first investigate the position of the tug.

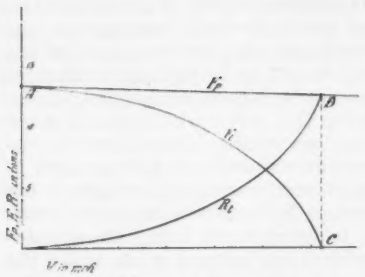


Diagram of forces acting on the tug.

The magnitude of F_p depends on the power and the efficiency of the engine and on the performance of the propelling equipment. It has its greatest value when the engine is developing maximum power while the tug remains forcibly fixed in place. As soon as the tug is let go, F_p diminishes at the rate of 2 to 4½ percent approximately for every mile per hour of increase in velocity, owing chiefly to the relieved expansion of exploded gases or steam behind the accelerated pistons or turbine blades. Concurrently, the resistance of the tug begins to increase. At the time when the ship has attained her maximum speed the resistance has also reached its highest mark, which means, that the propulsive force F_p is wholly spent in overcoming the resistance of the tug alone.

At that moment no force is available for towing purposes. Thus—

$$F_p = R_t \text{ and } F_t = O \quad . \quad . \quad (2)$$

The variations of these several forces are graphically illustrated in Fig. 1. Three interesting points are to be noted in the diagram, viz., the point A of maximum F_p and F_t, the point B of maximum R_t and the point C of maximum speed which the tug can attain with no load, coinciding with the minimum (zero) value of F_t. These points are characteristic of the tug's capacity to do work.

The difference at any moment in the magnitudes of F_p and R_t gives the force F_t remaining available for towing purposes. With the increase of speed the tractive force F_t diminishes, first because F_p decreases and then because R_t increases. Its variations follow a parabolic law. The change of F_t is of prime importance in our problem as it is the only useful force of all the forces involved; a good deal of economy of transport depends on its being cheaply produced and applied with good sense. And so there will be no harm if we dwell upon it a little longer.

THE WORK DONE BY THE TRACTIVE FORCE

Consider the tug moving at a certain speed; her tractive force is of potential character so long as there is no vessel in tow. But as soon as a barge is attached to the tow-line, the tractive force takes an active aspect and starts doing work, coincident wtih the reduction of the speed of the convoy. The work done by F_t in unit time is, mechanically, a product of $F_t \times V$, but in terms of useful work it is something else. Here we must draw distinction between these two aspects of mechanical work, viz., the work of traction measured in ton/miles, which the tractive force is actually doing to overcome the resistance of the barges

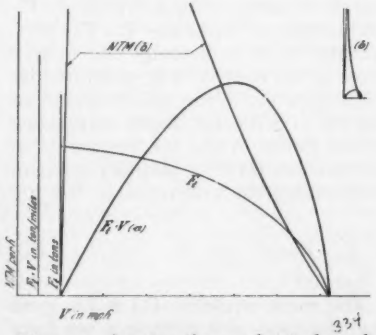


Diagram showing the mechanical work of the tractive force in its twofold aspect —$F_t \times V$ and NTM.

in tow, and the useful work reckoned in NTM (net ton/miles) which is merely a product of the useful load Q_b contained in barges, in tons, and the distance S in miles moved by them along the bank. See the curves $F_t \times V$ (a) and NTM (b) in Fig. 2.

The run of the curves shows clearly that, barring the points of zero value, the two kinds of work are obeying quite different laws, both as regards their magnitude and their structure. The origin of the useful work will be dealt with subsequently. In the meantime, the work of traction, for instance, is quite independent of the quantity of useful load Q_b and may even be performed with barges containing no load at all, e.g., during tow of empty barges. Conversely, the amount of useful work may be much more affected by some external factors—the strength of current for instance, the depth of the waterway, the form of the barges, the composition of the tow, etc.—than by the variations in tractive force. In its turn it can be performed with no tractive force, e.g., while drifting downstream with the velocity of the current. The point of greatest useful work is reached at infinitesimal speeds, much earlier than in the case of the work of traction and rather too soon; obviously there is no proportionality and no similarity between these two aspects of mechanical work except in their common trend to reach zero-value at relatively low speeds.

This is one of the main characteristics of the tractive force generated in a tug and but for the fact that other instruments of transport (excepting a seagoing ship) are more severely limited in that respect, the feature might be considered as her serious drawback.

Returning to the two kinds of work, no matter what inconsistencies might exist between them, one is the outcome of the other, that is, the useful work is the proper sequence of the work of traction. The useful work implies the presence of a useful load, of course, and since the tug contains no such load her sole function being seemingly the production of the tractive force with the work of traction as a consequential product, the metamorphosis of the latter into useful work occurs in all probability somewhere along the tow-line or rather at the end of it, the barges acting as intermediary. So we must focus our attention upon this second agent as well in order to grasp the events called forth by the tractive force, which are contained in the equation (1), viz., $F_t = R_b \times N$.

THE JOINT USEFUL WORK

Let the tow consist of a number of loaded barges of the following size and characteristics: L x B x T_{max} = 240 ft. x 29 ft. 6″ x 7 ft. 2″, D_{max} = 1205 t, Q_b = 1000 t. This is the type common on the Danube and the Rhine. Using Dr. Strickler's formula[2] for such vessels we get approximately the resistance of the barge in kg—

$$R_b = 40\ V^2 \quad . \quad . \quad . \quad (3)$$

where V is the speed of tow in statute miles per hour, relative to the water.

Subordinate to the magnitude of F_t, the number of barges N is obtained from the equation (1)—

$F_t = R_b \times N = 40\ V^2 \times N$, wherefrom

$$N = \frac{F_t}{40\ V^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The tow consisting of N barges with Q_b = 1000 t each, the total useful load will be:

$Q_t = 1000\ N$ in tons.

Substituting from (4)—

$$Q_t = 1000 \frac{F_t}{40\ V^2} = 25 \frac{F_t}{V^2}, \text{ wherefrom}$$

the useful work done per hour at the speed of V mph—

$$W = 25 \frac{F_t}{V^2} \times V = \frac{25\ F_t}{V} \text{ in NTM}^{(3)} \quad (5)$$

It follows from the above equation (5) graphically illustrated in Fig. 2, that the smaller the speed, the greater the useful work will be. This sounds improbable, yet on closer inspection of the change of F_t, as shown in the Fig. 1, we see, that the magnitude of the tractive force is inversely proportional to the speed of the tug, being greatest at V=O. At very low speeds, then, the tractive force is approaching its maximum value. It is also a matter of common knowledge that the resistance of barges at very low speeds is very small, so that the relation $F_t = R_b \times N$ *at very low speeds implies a very high N indeed.* Assuming, in fact, a tractive force of 5 tons at the speed of half a mile per hour, we get from the above relation (4)—

$$5000 = 40 \times (\tfrac{1}{2})^2 \times N,$$
$$N = 500 \text{ barges!}$$

But this fantastic number diminishes rapidly with the increase of speed, being approximately 120 at 1 mph, about 12 at 3 mph, about 7 at 4 mph and so on.

The amount of useful work in NTM changes as violently as the number of barges, of course, being 250.000 NTM at the speed of half a mile per hour, 57.500 NTM at 2 mph, 27.500 NTM at 4 mph and so on, which bears out the statement previously made concerning the post-separation trends of the tug.

In the light of what has already been said about the work of traction, its appearance may be looked upon merely as a phase process in the production of useful work, which is essentially a product of the tractive force F_t generated in the tug. The participation of barges in the making of NTM is, therefore, only apparent. Nevertheless, their imputation to the total cost of convoy is quite real and may not be neglected. And since the economic speed we are in search of is a component also of the total cost contracted by the convoy while doing useful work, our next objective will be the appraisal of such cost incurred by the tug and the barges respectively.

THE TOTAL COST PER NTM

We will suppose that the total annual expenditure to run a 1.200 HP tug amounts to $145,000.00 (The figure is purely tentative). Assuming 2,850 hours for useful work under full load during the same period, the cost of the tug per hour is about $51.00. For the sake of simplicity in subsequent arithmetical operations we may further assume, without committing an appreciable error thereby, that this cost is independent of the speed, which is right enough for the operative range of speeds.

The rate of production of NTM is shown by the curve (b) in Fig. 3. Dividing the cost per hour by the corresponding amounts of NTM realized at various speeds, the cost of NTM per tug is obtained. See the curve (c) in Fig. 3.

The cost of barges is computed in a similar way. Here, too, we shall assume the annual expenditure of same to be $8,400.00 per barge. (The figure is purely tentative.) Allowing 1900 hours for useful work at full load per annum, a cost of one barge per hour amounting to $4.40 is obtained.

It may now be recalled that the

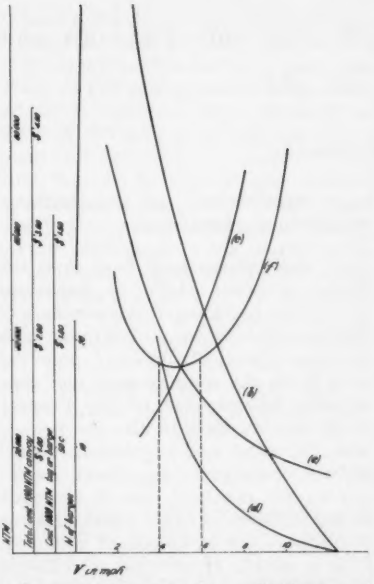


Diagram showing the individual and total cost of NTM incurred by the traction and tow components of the convoy.

number of barges required to create the full-load conditions for the tug varies with the speed of convoy in accordance with the equation $N = \frac{F_t}{40\ V^2}$, so that from point to point along the curve (b) the necessary number of barges is represented by the curve (d) in Fig. 3. Multiplying this number by 4.40 we get the cost per hour as the contribution of the barges to the total cost of performing useful work, and dividing the incident amount as before by the corresponding number of NTM at various speeds we obtain finally the cost of NTM for barges. See curve (e) in Fig. 3.

Obviously, the total cost of NTM is the sum of the costs contracted by the tug and the barges respectively. See curve (f) in Fig. 3.

We observe from the diagram that in our hypothetical case the minimum cost occurs at the speed of 5 mph, which is then, *the economic speed (Ve)*—of the convoy. As there is no appreciable difference in cost at the speeds of 4 and 6 mph we may consider these limits as the operative range of the economic speed.

The total economic load within the range varies from about 19,000 tons to 7,630 tons contained in 19, i.e., 8 barges, the economic static load per HP is varying from 15.8 tons to 6.4 tons at the speed of 4 and 6 mph respectively. See Table I.

TABLE I—*Of Computed Cost Per 1000 NTM-Convoy*

1. V in mph	4	5	6	7	8	10	12
2. F_t from the Fig. 1 in kg	12.200	11.800	11.300	10.600	9.400	6.500	2.000
3. NTM $= \frac{25}{V} F_t$ per h	76.200	59.000	47.000	37.800	29.400	16.250	4.170
4. Total cost tug per h, $				51.00			
5. Cost tug for 1000 NTM (4:3) $	0.680	0.863	1.085	1.350	1.734	3.120	12.250
6. N of barges $= \frac{F_t}{40\ V^2}$ (ea) ...	19.1	11.8	7.63	5.4	3.67	1.62	0.35
7. Cost per barge per h, $				4.40			
8. Total cost barges per h (6x7) $	84.00	51.80	33.60	23.80	16.15	7.13	1.54
9. Cost barges for 1000 NTM (8:3) $	1.106	0.878	0.735	0.628	0.548	0.438	0.369

10. Total cost convoy for 1000 NTM (5 + 9), $	1.786	1.741	1.820	1.978	2.282	3.558	12.619
11. Economic load in tons/HP (operative range of speeds)	15.8	10	6.7	..	..	..	..

THE ECONOMIC SPEED—Ve

The economic speed is the function of a large number of factors, most diverse in character and origin. Broadly speaking they can be listed under the following headings: the tug, the tow and the crew. Later on we shall have to add the waterway as well.

There is no question but that the tractive force generated in the tug is the factor number one on which the height of economic speed depends; the greater the F_t the higher the economic speed. But this is rather an incomplete statement which may lead to false assumptions. It is, in fact, not sufficient for the force F_t to be only large in order to involve a high economic speed; it must also be cheap in production and maintenance. Of the two tugs possessing identical traction characteristics the one with the lower running cost will claim a higher economic speed, all else remaining unaltered.

The tow, again, is a very important influence factor for the reason that it is the consumer of the tractive force; it may be a sparing or an extravagant consumer. If the tow be of such a pattern and composition that its specific resistance—hence the specific consumption of F_t—is low, the economic speed will be high, and vice versa. That is why the combination of barges intended for push-towing allows of a higher economic speed than the pull-tow formation. In addition to the influence of the type of tow, the size and shape of the barges also affect the economic speed. A large-sized, streamlined barge would promote a higher economic speed; a small sized vessel or a wooden hull, on the contrary, would claim a lower one.

As everywhere else here, too, the human element plays an important part in the handling of this problem of traction. Hereby we do not mean only the expert skill of the crew, essential as it is, in the maintenance and reasonable consumption of the tractive force, but we include also the principles, the rules and regulations issued by the operator-management, determining the practical line of approach to the problem of rapid transportation of bulk-freight by means of a convoy. It is a matter of common knowledge that high speed transport on sheltered waters is rather expensive; on the other hand, speed is the fundamental condition of our way of living today. And so it will call for some hard, clear thinking and sound decisions on the part of the human component to associate it with the economy.

From the scanty choice of pertinent information supplied by the American periodicals (coming our way), it may be concluded, that the operative speed of convoys plying the Mississippi-Ohio river system is generally in excess of the one computed as shown and termed as economic. The recent acquisition of the *Ashland* fleet,[4] for instance, the m.v. *Allied-Ashland* (3 x 1600 HP) is designed for a speed of 9 mph in still water with a load of 24,000 tons in 8 barges. Now we admit being rather handicapped by the lack of knowledge of much data concerning the tug and the barges, indispensable for a comprehensive computation of the economic speed for such a convoy, but appraising carefully the possible traction

capacity of both components and evaluating, as best we can, the missing links in our calculations, the economic speed of this convoy seems to be nearer to 6½ than to 9 mph. However, in view of the alleged advantages of the push-towing system (sparing consumption of F_t due to the reduced resistance), a speed of 7⅓ mph might easily prove to be economic; finally, giving preference to the higher value of economic speed range one could very probably contend the operative speed of 8 mph to be also economic. This would certainly be more profitable than to travel with the velocity corresponding to the lower value of speed range. At such a speed, then, the tug A-A could push 11 barges instead of 8, the difference in useful load being about 9000 tons. Or else, the tow of 8 barges could be pushed at the speed of 8 mph by a tug developing approximately 22 percent less horsepower, the last mile being ever the most difficult to overcome.

Nevertheless the cost of NTM at 9 mph being about 13 percent higher than at 8 mph, one might suppose that the operator-management see their provisory advantage in the increase of operative speed rather than in the enlargement of dumb fleet with the high initial cost it would require.

TOWING UPSTREAM

Hitherto we have been dealing with the traction in still water,—rather a venture in theories. With the current C to be overcome, the realization of NTM is delayed, the speed of advance of the convoy being now (V − C) along the bank where C is the speed of current in mph.

The equation (5) takes the form of—

$$W = 25\frac{F_t}{V^2}(V-C) \quad . \quad (6)$$

for barges with $Q_b = 1000$ t.

Using the equation for different values of C, various curves (G) of the useful work in NTM are obtained. The subsequent process of computing the cost of NTM per tug is identical with the one used previously. As the speed of advance is now lower than before (though the ship is developing again the full power), resulting in a smaller supply of realized NTM for the same amount of total expenditure, the cost of NTM is, naturally, higher than before; it increases with the current, naturally. See Fig. 4.

The cost per NTM per barge is found accordingly or we could imagine the ostensible making of NTM by this component at the rate of (V − C) per hour and divide the cost per hour of one barge by the NTM thus made, adding finally the quotient to the cost of NTM per tug, in order to obtain the total cost of NTM-convoy.

The apparent realization of NTM per barge, be it mentioned by the way, follows the straight line law of strict proportionality; a number of NTM will be made at the speed of V mph, while at the speed of 2V, the amount of useful work will exactly be doubled, the cost of NTM diminishing asymptotically. Here may be noticed the trend of the barge to work most economically at the highest possible speed.

While advancing against the current C the process of making NTM goes on precisely as in still water, but with a retardation of the magnitude equalling the strength of the current. See straight lines (f) in Fig. 4.

The joint (total) cost of NTM rises with the current geometrically. The freight rates are soon doubled. See curves (h) in Fig. 4.

The economic speed also increases, but in a less violent manner.

THE TRAVEL DOWNSTREAM

The application of formula (6) now put in adequate form to meet the changed conditions, i.e.—

$$W=25\frac{F_t}{V^2}\ (V+C) \quad . \quad (7)$$

will furnish the correct answer on the score of economic speed; correct, that is, algebraically but rather baffling otherwise, as it implies with the increasing speed of current an ever diminishing operative speed of the convoy. The economic speed of tow downstream is lower than in still water, which is generally speaking low enough as it is; it is approximately, but not quite, equal to the difference between the latter and the speed of current. To fare economically would mean, therefore, for the tug to develop a low technical speed, by which is understood the speed accomplished through the exertion of the technical equipment aboard the tug.

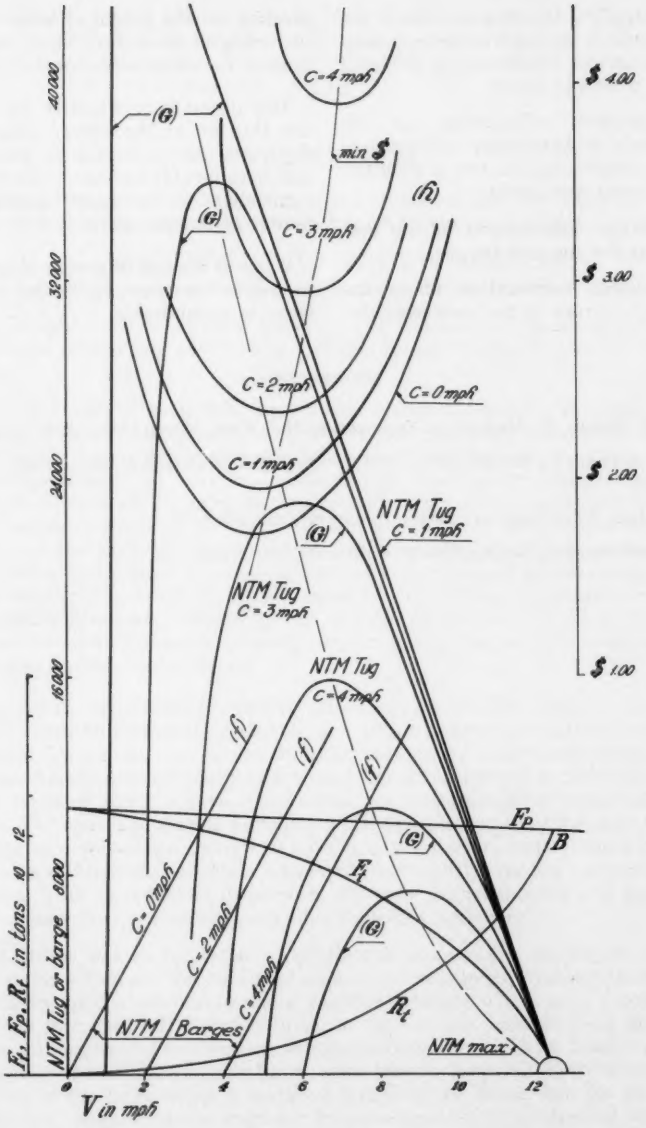
Now there are two possible basic conditions for a highly powered towboat to advance slowly downstream: either to handle a tow consisting of an excessive number of barges or else, to reduce the power. The first condition will most certainly shatter through the practical difficulties, while the alternative, properly speaking, is not an economical proceeding. Leaving aside the implied contrariness of the first proposition to achieve the economical transport by retarding the movement of the barges beyond the limits set down for the economic speed in still water regime, we are confronted in the alternative with the task of reducing the freight-rates by the application of methods distinctly uneconomical.

Obviously the equation (7) offers no solution to the problem of economic speed downstream. In reality, the problem is more of the routine than of mechanics. In the latter aspect it can easily be cleared away if we treat the convoy as subjected to a complex motion combined of the movement caused by the engine plus the motion of the ambient water. The speed developed by the engine should be equal to the speed of convoy in still water (coincident with the technical speed), while the economic speed downstream will now be the sum of the former and the speed of the current. If the actual circumstances would allow such an arrangement to be made, one could contend that the transport of cargo is being performed economically; if not, one will have to proceed according to the needs of the moment and the place.

CONCLUSION

The economy of transport by means of convoy on inland waterways rests on three main assumptions, viz: the cheap production and sensible application of the tractive force generated by the tug, the full use of this force for the longest possible period and the economic speed of tow. The realization of the first supposition is the matter of joint concern of mechanical engineer and naval architect when designing the components of the convey; the long-term use of the force F_t is best insured by the preliminary planning of transport, while for the transportation at economic speed a thorough knowledge of the tractive characteristics of tug and barges, as also of hydrologic properties of the waterways is indispensable.

To that end the tug ought to be provided with DISPATCHER CHARTS which must contain:



Characteristic curves of NTM at various speeds of advance against a variable current C for the convoy consisting of a 1200 HP tug and 1000 t barges.

1) complete information about the magnitude of the tractive force in most diverse service conditions, at different power, draft and speed;

2) complete information on the magnitude of individual resistance of barges constituting the tow at different load (draft) and speed;

3) precise information of the cost per hour for tug and barges;

4) reliable information about the speed of current of the waterway depending on the height of water-level, including all other data which may influence the composition of the convoy.

The dispatcher, whether on board the ship or in the office, where the duplicate charts should be available, will then be able to form in all the circumstances an "economic" convoy as a matter of routine work.

Without the use of proper charts the economic transport on inland waterways is unthinkable.

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"PERMOBILITY"

"A New Criteria for Arrangement of Naval Machinery," by John H. Wright, published in Volume 66, Number 3, August 1954 of the Society's JOURNAL.

A COMMENT BY DAVID J. BARRY

(For biographical sketch see Volume 64, No. 4, November 1952)

Mr. Wright's article, of basic interest in all aspects of ship arrangement, has coined the new term "permobility." It will be interesting to observe if it will survive.

This permobility to which the author wishes to direct some constructive thinking, has as a background for analysis, the machinery arrangements which were developed for the large battleships and aircraft carriers. Perhaps the radical compartmentation that was made for the machinery arrangement of the CVA-41 Class could be taken as the utmost in permobility up to that time. The large number of machinery compartments used in that design to achieve separation of the main propulsion plant was questioned from an access point of view. Since this degree of isolation has not been repeated in later designs, it is to be assumed that such an arrangement was more than adequate to meet the permobility standards desired by Mr. Wright. At the time of the design, there was considerable discussion as to the advantages of the multiple room arrangement that was finally developed.

The author has, in the interest of simplicity, limited the scope of his analysis to ships with four propulsion shafts, each with a minimum of 40,000 horsepower per shaft. To the naval architect, these limitations are considered too restrictive. The machinery arrangements and disposition of machinery, a problem common to all types of ships, is even more critical on ships having a lesser number of shafts. The spacings of main sub-division bulkheads required for survival after damage in a ship large enough to have four propulsion shafts, would in themselves be sufficient to provide a degree of "permobility" for the machinery components. Thus is achieved dispersion, not only for machinery and piping but also for personnel, armaments and other vital ship functions.

The armor box in capital ships of the size discussed in the article, contains those items which are considered of importance to ship survival and includes the machinery spaces, ammunition and gasoline stowage. There is so much weight involved that the hull designer must, at the earliest possible time, obtain the minimum length and fore and aft location to develop the ships lines. In defense of the naval architect, it is only fair to state that in the preliminary arrangement of many of the large ships, a tentative length in the armor box for machinery spaces was assigned to the engineer for development of the desired machinery compartmentation.

This writer is inclined to disagree with Mr. Wright's statement that "there are no universally accepted naval machinery arrangement criteria." Although

the rules have not been formally stated, the underlying universally accepted arrangement criteria has always been to disperse and to segregate main components in order to minimize damage from explosion.

It would appear to be self evident that the permobility problem on a large multiple propeller ship without side protection is of a more complex nature than on that of a ship with heavy side protection. The torpedo protection tends to minimize the length of hit and to restrict the fore and aft flooding caused by such damage. For this reason, the permobility of an unprotected hull has to be closely associated with the damage control features of the ship for survival after damage. Conversely, with adequate torpedo protection, permobility could be a study of machinery arrangement only.

In this connection, the article provoked thoughts about additional permobility studies which could be made. For example, could side protection be minimized by designing permobility to a predetermined standard such as the length of damage expected from a torpedo hit? If a saving in the weight of side protection could be achieved thereby, the result would be beneficial to the design. In the same manner, the subtraction of the weight of a transverse bulkhead in the armor box would also be a considerable saving in weight. Thus, in any discussion on criteria for machinery compartmentation, it appears that each machinery arrangement requires special consideration and integration into the design of the hull structure. The complete ship to be in agreement with the basic criteria for permobility, viz., "keep them isolated, keep them short, and keep them separated one from another."

Assuming that the machinery problems can be assessed according to good permobility criteria, there are many other areas of possible damage which could also be studied. For example, in the types of ships to which Mr. Wright limits his discussion, the shaft alleys and shafting extend fore and aft for many hundreds of feet. Because of this condition, there is potential damage to multiple machinery components in many locations over a long hit area of the hull. In addition, as the shafts go aft, they overlap each other in a broad side arrangement. This then, is another permobility problem for solution because a hit could cause multiple propulsion breakdowns in space far removed from the actual location of the damage. This possibility leads to conjecture as to how better permobility could be achieved for shafting. It will require advance thinking on the part of the machinery designers and naval architects. For example, is it conceivable that the hull could be effectively formed to permit installation of special type propellers distributed throughout the length of the ship, or could the propellers be changed in type? The principal benefit of such an arrangement would be in the elimination of most of the shafts and in the structure for the shaft alleys. This saving in space and weight would be appreciable and might compensate in part for the radical development necessary to locate the propellers closer to their driving machinery. The above does not offer any new thought and is cited merely to emphasize that the problems concerning the permobility in machinery spaces are tied up with those of other functions and component parts of the ship. While the disposition of the machinery spaces and the location of the auxiliary components have always been factors of design in existing U. S. Naval Vessels, Mr. Wright has performed a thoughtful service by his emphasis on this subject. This article presents this problem in writing for the first time. It should be of great value for training purposes and in the design of naval and auxiliary vessels.

STRENGTH AND RESISTANCE OF METALS

By JOHN M. LESSELLS
Published in 1954
John Wiley & Sons, Inc. and Chapman & Hall, Ltd.
440 Fourth Avenue, New York 16, New York
450 pages with Illustrations
Reviewed by ASST. PROF. L. R. MANN,
Department of Marine Engineering, U. S. Naval Academy

In a unique fashion, while readily readable, a text is finally available which ties the work of the engineer and the metallurgist together. Concerning itself with the basic engineering material, steel, the book is aimed at the graduate student and practicing engineer with its purpose being a study of the behavior of metals under elastic stress. However, mention is made of non-ferrous alloys and cast iron in those particular instances where their behavior differs from that of steel.

Bringing to the text many years of experience in the field, Associate Professor Emeritus (M.I.T.) Lessells, now head of his own engineering company, takes each of the basic types of stress and reviews their physical testing methods in the light of the most recent investigations.

Throughout the book, each subject is authenticated with tabular data from results of many and varied tests. Specific reference is made in each chapter to the major contributions by important workers in their fields, which gives a very "human" approach to the subject.

After reviewing rudimentary testing procedures in the first four chapters, Professor Lessells goes into great detail in the next five chapters on the highly important subjects of impact and fatigue.

The adoption of higher temperatures and pressures in modern steam turbines, together with the development of the gas turbine make the behavior of metals at high temperature of extreme importance to the engineer. Chapter 3 discusses briefly the short-time, rupture, and creep tests of carbon and alloy steels as well as gas-turbine superalloys, and it reviews the method whereby such test data may be used in design.

Economic value, both of material and research, is judiciously examined throughout the text. Chapter 10 contains a discussion of a cause of great economic loss, mechanical wear, together with the classic theories of friction.

The last chapter in the book discusses a subject dear to the hearts of all practicing engineers—the subject of working stresses. The various influences governing the choice of safety factors for proper working-stress evaluations are included. Since the final choice of working stresses depends on the mechanical properties of the material employed, tables of such properties for the metals and alloys in common use are included.

The text has problems in it on the subject matter of each chapter, but is primarily intended as a source book rather than a textbook in the opinion of this reviewer.

Rear Admiral ALBERT THOMAS CHURCH, USN, Retired

OBITUARY

Albert Thomas Church

On November 14, 1954, in Annapolis, Md., Rear Admiral Albert Thomas Church, U.S. Navy, retired, died at the age of 72.

Admiral Church graduated from the U.S. Naval Academy in 1905 and most of his active naval career was devoted to Naval Engineering. During tours of duty as Repair Officer of destroyers operating from British bases during World War I, as head of the Navy Postgraduate School from 1927 to 1931, as manager of the Navy Yard at Bremerton, Wash., and as Director of the U.S. Naval Engineering Experiment Station during World War II, he was an important contributor to the advance of Naval Engineering.

A long time member of the Society, Admiral Church served as Secretary-Treasurer in 1915 and 1916.

DIED

It is with the utmost regret that the Society has to report the receipt of the notice of the death of the following since the publication of the November, 1954, Journal.

BERTRAND, ERNEST R., Naval Member

CHURCH, ALBERT T., Naval Member

JOHNSON, WARREN TRUMBULL, Civil Member

JOHNSON, WARREN W., Naval Member

ROBERTSON, WALTER W., Civil Member

WILLIAMS, D. A., Civil Member

CHANGES IN MEMBERSHIP

The Society announces with much pleasure that the following have joined as members since the publication of the November, 1954, Journal:

NAVAL

Blick, Gustavus Worthington, Commander, USNR
Chauncey Place, Mt. Rainier, Md.

Bortner, James Augustine, Lieutenant, jg, USNR
USN Administration Unit, MIT, Cambridge 39, Mass.
Mail: 286 Gray St., Arlington 74, Mass.

Hoffman, Charles M. E., Rear Admiral, USNR, Ret.
3420 Guilford Terrace, Baltimore 18, Md.

Leete, Charles Ernest, Commander, USNR
Assistant to President, Bogue Electric Co.
Mail: 402 Benson Drive, Alexandria, Va.

Signell, Warren Ivar, Lieutenant, USNR, Marine Engineer
Foster Wheeler Corp., 165 Broadway, New York, N.Y.

Smalzel, Charles Warren, Commander, USN
Head Applied Science Branch, Code 370, Bureau of Ships
Navy Dept., Washington 25, D.C.

Sprung, Emmett Emerson, Captain, USN, Ret.
Engineering Representative, Vacu Blost Co., Belmont, Calif.
Mail: 237 Prince George St., Annapolis, Md.

Thompson, Ralph F. J., Lieutenant, USNR
Assistant Manager, Federal Sales, Reynolds Metals Co.,
503 World Center Building, Washington, D.C.

Waugh, John Robert, Jr., Lieutenant Commander, USMC, Ret.
Sales Manager, Washington Office, Lewyt Mfg. Corp.
60 Broadway, Brooklyn, N.Y.
Mail: Hotel Dupont Plaza, Washington, D.C.

CIVIL

Boger, Albert R., Buyer, Newport News S.B. & D.D. Co.
Mail: 105 35th St., Apt. 5, Newport News, Va.

Conover, George Wallace, Jr., Dist. Manager, Scintilla Div.
Bendix Aviation Div., 1701 K St., NW, Washington, D.C.

Daman, Ernest Ludwig, Deputy Director of Research
Foster Wheeler Corp., 165 Broadway, New York, N.Y.
Mail: 532 Coleman Place, Westfield, N.J.

Ewalt, Newton C., Manager Power Plant Division
Walworth Co., 60 East 42d St., New York 17, N.Y.

Howard, Allen Waller, Sales Engineer,
General Electric Co., Wyatt Bldg.
777 14th St., NW, Washington, D.C.

Kosak, Rudolph J., General Engineer, Design Section
Supt. of Shipbuilding
Mail: 4173 Birmingham Road, Jacksonville, Fla.

ASSOCIATE

Belby, Michael, Electronics Engineer, General Electric Co.
Mail: RD No. 1, Paris Hill Road, Sauquoit, N.Y.

Bingham, James Joseph, Gas Turbine Specialist
General Electric Co., Suite 1014 Wyatt Bldg.
777 14th St., NW, Washington, D.C.

Giacobbe, Eros, Marine Engineer
Via Penisola 12, Genoa, Italy

Hanson, Peter B., Sales Representative
U. S. Steel Corp., 1025 K St., NW, Washington 6, D.C.

Humphreys, William Thomas, Assistant to Manager, Fairbanks Morse & Co.
Mail: 6644 Hillandale Road, Chevy Chase 15, Md.

Kinsey, Lloyd F., Engineer
Burroughs Corp., 1739 H St., NW, Washington, D.C.

Spandau, Irwin B., Industrial Engineer
1290 E. 19th St., Brooklyn 30, N.Y.

RESIGNED

Naval

Allen, Charles D.
Armstrong, Charles V.
Armstrong, Walter B.
Bradford, L. M.
Bryant, Eliot
Caskey, M. W.
Clark, Lloyd James
Davis, Ivor F.
Dunston, Whitney A.
Goky, Noah W.
Green, Samuel W.
Helber, Carlyle L.
Herbert, Clarence E.
Hopkins, John Ray
Jones, C. E.
McDevitt, John Joseph
Macaulay, W. S.
Moale, Edward S.
Morser, Calvin S.
Roach, Richard T.
Roberts, James M.
Row, Charles Rockwell
Shores, Robert M.
Williams, W. B.

Civil

Blackwood, W. J.
Drennan, Thomas W.
Engel, Emil W.
Lincoln, Joseph B.
Lindgren, John K.
Nichols, F. C.
Robinson, Fielding S.
Saunders, L. P.
Spofford, Wolcott E.
Stevens, E. A., Jr.
Taylor, S. J.

DROPPED FROM ROLLS

In accordance with the requirements of the By-Laws, it was necessary to on 31 December 1954, to drop from our rolls 109 members who were two years in arrears for dues.

ASSOCIATION NOTES

ORIGINAL ARTICLES

The editor of the JOURNAL is most anxious to obtain manuscripts of original articles on subjects of interest to naval engineers. Any manuscript accepted for publication becomes the property of the Society which copyrights it upon publication. Authors are paid from $50.00 to $250.00 depending upon length, interest and professional value.

The rules for manuscripts are as simple as we can make them.

1. They must be legible (typewritten, double-spaced preferred) since no proof is submitted to authors for correction prior to printing.

2. Single copy only is required.

3. Mathematical formulae used must be set up, preferably by hand printing, so that there can be no question as to symbols, subscripts and exponents.

4. Any drawing submitted must be black on white. A cloth or paper tracing is acceptable but not required.

5. Photographs may be negatives, but glossy prints are acceptable.

6. Include on a separate page a short biographical sketch(es) of the author(s). 50 to 100 words for each author is desired.

Manuscripts should be addressed to:

Secretary-Treasurer
American Society of Naval Engineers
605 F St., N. W.
Washington 4, D. C.

Manuscripts accepted will not be returned unless specifically requested by the author. If returned, they will be in the condition which has resulted from the work of the printer and the engraver. Immediately following publication, the author is furnished 10 reprint copies of his article free of charge. Additional copies may be purchased from the Society if the request is received 30 days prior to the publication date which is the 25th of the issue month. Estimate of cost of additional reprints, which will vary with the nature of the article and the number of copies ordered, will be furnished on request as soon as possible after the article is set up.

Manuscripts not accepted will be returned, postpaid, by the Society.

Manuscripts which are accepted will be published in the earliest JOURNAL which has not yet been closed (60 days before publication) and for which sufficient material is not already on hand.

PHOTOGRAPHS

Photographs of current or historical interest to readers of the JOURNAL are desired. Any which are accepted by the Society will be purchased by the Society at a standard price of $5.00 each. Such purchase *will not* include any copyright by the Society, but the contributor must hold the Society free from any charge of violation of any previous copyright.

NOTICE OF DEATHS

The Society has no satisfactory machinery for obtaining notice of deaths of members. It is particularly desired that we receive such notice upon the death of any past officer, past member of the Council or member of more than twenty years' standing with photographs and short obituaries. It will be appreciated greatly if any member who learns of the death of a member will advise the Secretary-Treasurer.

SOCIETY LAPEL BUTTONS

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold background.

It is available to all members at fifty cents (50c) each.

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> "The views expressed herein are the personal opinions of the author and are not necessarily the official views of the Department of Defense or of a Military Department."

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Captain J. E. HAMILTON, U. S. Navy, Retired

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1891	Assistant Engineer EMIL THEISS, U. S. Navy
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